TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 47

FORTY-SEVENTH ANNUAL MEETING
KANSAS CITY, MO., JANUARY 27-29, 1941

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SEMI-ANNUAL MEETING
SAN FRANCISCO, CALIF., JUNE 16-20, 1941

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PUBLISHED BY THE SOCIETY AT THE OFFICE OF THE SECRETARY
51 MADISON AVENUE
NEW YORK, N. Y., U. S. A.

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Secretary, T. F. Rockwell
Treasurer, L. S. Maehling
Board of Governors: F. C. McIntosh, H. B. Steggall, G. G. Waters

St. Louis

Headquarters, St. Louis, Mo.
Meets: First Tuesday

President, D. J. Fagin
First Vice-President, M. F. Carlock
Second Vice-President, C. F. Boester
Secretary, J. H. Carter
Treasurer, W. J. Oonk
Board of Governors: L. J. DuBois, C. E. Hartwein, B. C. Simons, J. S. Rosebrough

South Texas

Headquarters, Houston, Tex.
Meets: Third Friday

President, D. S. Cooper
Vice-President, R. M. Spencer
Secretary, A. M. Chase
Treasurer, L. S. Pawkett
Board of Governors: A. B. Banowsky, W. R. Etie, A. J. Rummel

Southern California

Headquarters, Los Angeles, Calif.
Meets: Second Wednesday

President, A. J. Hess
Vice-President, H. H. Bullock
Secretary, Leo Hungerford
Treasurer, W. O. Stewart
Board of Governors: H. H. Douglas, J. B. Griffith, Maron Kennedy, J. F. Park

Washington, D. C.

Headquarters, Washington, D. C.
Meets: Second Wednesday

President, F. A. Leser
Vice-President, R. S. Dill
Secretary, F. M. Thuney
Treasurer, K. F. Baldwin, Jr.
Board of Governors: Rutcher Skagerberg, F. E. Spurney, A. E. Stack

Western Michigan

Headquarters, Grand Rapids, Mich.
Meets: Second Monday

President, W. G. Schlichting
Vice-President, F. C. Warren
Secretary, H. J. Metzger
Treasurer, A. H. Snook
Board of Governors: H. D. Bratt, C. H. Pesterfield, T. D. Stafford

Western New York

Headquarters, Buffalo, N. Y.
Meets: Second Monday

President, W. R. Heath
First Vice-President, H. C. Schafer
Second Vice-President, S. M. Quackenbush
Secretary, Herman Seelbach, Jr.
Treasurer, B. C. Candee
Board of Governors: M. C. Beman, Joseph Davis, Roswell Farnham, C. W. Farrar

Wisconsin

Headquarters, Milwaukee, Wis.
Meets: Third Monday

President, T. M. Hughey
Vice-President, W. A. Ouweneel
Secretary, O. A. Trostel
Treasurer, H. W. Schreiber
Board of Governors: A. S. Krenz, D. W. Nelson, Ernest Szekely

XUM

TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1168

FORTY-SEVENTH ANNUAL MEETING, 1941

Kansas City, Mo.

KANSAS CITY, MO., welcomed the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS on its 47th Annual Meeting, January 27-29, 1941, at the Hotel Muehlebach, with an attendance beyond expectation, including 246 members, 101 guests and 71 ladies, a total of 418 registered. The Council held a meeting during the afternoon of the 26th and in the evening there was a meeting of the Committee on Research. Eleven Technical Advisory Committees had meetings and two code committees held conferences. During the four technical sessions 15 papers were presented and discussed. Amendments to the By-Laws affecting future dues rates were presented. Delegates from 30 of the 31 chapters of the Society were in attendance at the Chapter Delegates Conference Monday morning.

The first session of the 47th Annual Meeting of the Society was called to order on Monday, January 27 at 2:00 p.m. by Pres. F. E. Giesecke in the ballroom of the Hotel Muehlebach, Kansas City, Mo. President Giesecke announced that the papers would be presented first and the reports would follow.

President Giesecke, following the presentation of the technical papers, called for the Report of the Tellers, which was submitted by C. S. Pabst, New York, as follows:

Report of Tellers of Election

BALLOT—OFFICERS

	For
President—W. L. Fleisher	640
First Vice-President—E. O. Eastwood	673
Second Vice-President—J. H. Walker	673
Treasurer—M. F. Blankin	673

Scattering votes: For M. C. Beman, H. W. Fiedler, E. O. Eastwood for *President;* for J. H. Walker for *First Vice-President;* for J. F. Collins, Jr. for *Treasurer;* and for N. D. Adams for *Member of the Council*

Members of the Council—Three-Year Term

A. P. Kratz	674
W. A. Russell	672
L. P. Saunders	675
C. Tasker	675
Total Ballots Counted	683
Total Ballots Legal	675
Ballots Invalid	8

BALLOT—COMMITTEE ON RESEARCH

Three-Year Term

	For
C. M. Ashley	675
M. K. Fahnestock	675
H. King McCain	675
F. C. McIntosh	674
T. H. Urdahl	674
Total Ballots Counted	683
Total Ballots Legal	675
Ballots Invalid	8

BOARD OF TELLERS

C. S. PABST, *Chairman*
C. H. B. HOTCHKISS, J. F. KERN, JR.

President Giesecke then called for the Report of the Finance Committee which was given by E. N. McDonnell in the absence of A. J. Offner, as follows:

Report of the Finance Committee

The following is the financial report for the year of 1940. As last year, your Finance Committee is again pleased to report Society finances as sound and in an excellent condition.

As required by the By-Laws, the accounts of the Society have been audited by a designated certified Public Accountant. The balance sheet and budget comparison prepared by Tusa & La Bella are a part of this report.

The balance sheet for 1940 shows that the net assets of the Society, including Research, amount to $152,943.76; of this total there is $68,960.94 in the operating and reserve accounts and $26,510.48 in the Endowment Funds in cash or invested in securities of the United States Government. Comparing the total income and expenditure of the Society during the past year with the estimated figures as set up in the Budget adopted by the Council at the start of the year, it is found that the income was in excess of the estimate by $5,569.62, while expenses were $1,559.06 less than Budget provisions. The operation of general Society activities shows a loss for the year of $147.13, but with the surplus from THE GUIDE and editorial contracts after deduction of reasonable reserves, leaves an addition to the General Fund amounting to $7,128.68. Research activities had an excess of $553.61 in income over current expense.

Subject to the requirements of the Constitution, 40 per cent of the dues, collected from Members and Associates and amounting to $16,402.20 was turned over to the Committee on Research. In addition to this amount the Council appropriated $5,000.00 making a total of $21,402.20 contributed by the Society during 1940 for Research work. The total amount spent in Research during 1940 was $51,386.67 which amount, in addition to Society contributions, also includes earmarked and unearmarked contributions and contracts. A detailed financial statement of the Committee on Research will be presented by the Chairman of that Committee.

During the past year the Finance Committee, with Council approval, has invested $7,500.00, the maximum permitted by law, in U. S. Savings Bonds of $10,000.00 par value at maturity date. This was added to the Endowment Fund not only to strengthen this fund but also to bring it closer to Constitution requirements.

Your Committee has during the past year carefully watched the income and soundness of our present investments. It recommends that surplus funds be retained in Savings Bank accounts, pending an opportune time for sound and legal investments.

Respectfully submitted,

FINANCE COMMITTEE,
A. J. OFFNER, *Chairman*
E. O. EASTWOOD, E. N. MCDONNELL

TUSA & LA BELLA
Certified Public Accountants
55 Liberty St., New York

January 20, 1941

American Society of Heating and
Ventilating Engineers,
51 Madison Ave.,
New York, N. Y.
Gentlemen:

Pursuant to your request, we made an examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y. and the Research Fund for the calendar year ended December 31, 1940 and submit herewith our report.

The work covered a verification of the Assets and Liabilities as of the date previously stated and a review of the operating accounts for the year ended December 31, 1940. For the period recorded cash receipts were traced into the depositories; the cancelled bank vouchers were inspected and compared with the cash records and the dues income was accounted for.

Submitted herewith is a balance sheet showing the financial condition of the Society on December 31, 1940 and your attention is directed to the following comments thereon:

CASH

Cash on Deposit was verified by direct communication with the banks and reconcilement of the amounts reported to us with the balances shown by the books of the Society.

The petty cash kept in the New York office was verified by count.

MARKETABLE SECURITIES

There is attached hereto a schedule of negotiable bonds which were verified by direct communication with the Bankers Trust Company where same are deposited for safe-keeping. No adjustment has been made of the $3,192.71 shrinkage in the market value of these securities, same being included in the attached Balance Sheet at cost.

ACCOUNTS RECEIVABLE

A list of the Membership Dues Receivable as of December 31, 1940 furnished to us by the management was checked to the individual ledger cards and found in

agreement with the general ledger control. The unpaid dues may be summarized as follows:

Dues billed in 1940	$ 6,695.12
Dues billed in 1939	401.45
Dues billed in prior years	3,943.56
Total	$11,040.13

Amounts due from GUIDE advertisers and other accounts Receivable were verified by trial balance of the individual ledger accounts which were found in agreement with the general ledger control.

After charging off $5,618.69 in uncollectible dues per Council authorization we have increased the reserve for dues doubtful of collection by $1,394.14. The adjusted reserves now shown on the attached balance sheet in our opinion are ample to cover any probable losses from future realization.

INVENTORIES

Items appearing under this caption are based upon quantities submitted by the management and computations made by us. The TRANSACTIONS inventory priced at cost may be summarized as follows:

Prior years	$1,750.00
1935–1936	70.95
1937–1938	377.62
1939	604.24
Total	$2,802.81

FURNITURE, FIXTURES AND LIBRARY

Furniture, Fixtures and Library are shown herein at the book values without appraisal by us; we did, however, provide for depreciation of furniture and fixtures at the rate of ten per cent (10%) per annum.

ACCOUNTS PAYABLE

On December 31, 1940 there remained unpaid invoices amounting to $11,219.29 which included the sum of $10,403.70 estimated by the management necessary to complete and make the first mailing of the 1941 GUIDE.

PAYABLE TO RESEARCH FUND

The amounts due the research fund representing 40% of Members and Associates dues were determined from computations made by us in accordance with *Section* 5, Article 3 of the By-Laws.

ACCRUED ACCOUNTS

Additional compensation to the secretary and the clerical staff of the Society has been computed in accordance with the instructions of the Finance Committee.

DEFERRED INCOME

Members' prepaid dues were ascertained by trial balance of the individual ledger cards. Prepaid dues from proposed members were verified from inspection of the applications found on file.

GENERAL FUND

There is attached hereto a schedule showing the changes that have occurred in the General Fund during the calendar year 1940.

SPECIFIC FUNDS

The specific funds are shown in the attached Balance Sheet after adjustment for interest earned.

The Balance Sheet and schedules showing the operations of the Research Fund are annexed hereto, also separate report has been prepared and rendered to the Committee on Research.

OPERATIONS

There are attached hereto statements of Income and Expenses for the calendar year 1940 showing an excess of Expenses over Income from Society activities of $147.13 and a net income of $7,275.81 from GUIDE operations. In the preparation of these statements 30 per cent of the salaries and office expenses have been allocated to GUIDE operations.

Respectfully submitted,
TUSA & LA BELLA,
Certified Public Accountants.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(December 31, 1940)

ASSETS

SOCIETY			
CASH			
On deposit		$37,999.95	
On hand		100.00	
Out for collection		90.00	
In closed banks		418.97	$ 38,608.92
RESERVE FUNDS			
On deposit			$ 11,169.29
INVESTMENTS (AT COST)			
Securities (Market value $19,479.17)		$19,178.23	
Add: Accrued interest		4.50	$ 19,182.73
ACCOUNTS RECEIVABLE			
Membership dues	$11,040.13		
Less: Reserve for doubtful	7,572.14	$ 3,467.99	
Advertisers and sundry debtors	$31,665.51		
Less: Reserve for doubtful	1,291.37	30,374.14	33,842.13
INVENTORIES			
Transactions—Copies		$ 2,802.81	
Transactions—Paper		548.00	
Emblems and certificate frames		71.14	
Stationery		39.30	$ 3,461.25
PERMANENT			
Library		$ 300.00	
Furniture and fixtures	$ 2,321.71		
Less: Reserve for depreciation	700.19	1,621.52	1,921.52
TOTAL ASSETS			$108,185.84
SPECIFIC FUNDS			
ENDOWMENT FUND			
Cash on deposit		$ 6,772.14	
Securities at cost (market value $16,133.34)	$19,626.99		
Add: Accrued interest	111.35	$19,738.34	$ 26,510.48
F. PAUL ANDERSON AWARD FUND			
Cash on deposit			1,145.82
			$135,842.14

LIABILITIES AND GENERAL FUND

SOCIETY	
ACCOUNTS PAYABLE	$11,219.29
PAYABLE TO RESEARCH FUND	
On dues as and when collected	4,232.77
ACCRUED ACCOUNTS	
Compensation—Secretary and staff	3,252.55
RESERVE FOR PUBLICATION	
Transactions—Volume 46	3,900.00

DEFERRED INCOME			
Prepaid dues—Members	$ 444.35		
Prepaid dues—Proposed members	244.72	$ 689.07	
TOTAL LIABILITIES			$ 23,293.68
GENERAL FUND			
Society			$ 84,892.16
TOTAL LIABILITIES AND GENERAL FUND			$108,185.84
SPECIFIC FUNDS			
ENDOWMENT FUND			
Principal		$24,063.28	
Unexpended income		2,447.20	$ 26,510.48
F. PAUL ANDERSON AWARD FUND			
Principal		$ 1,000.00	
Unexpended income		145.82	$ 1,145.82
			$135,842.14

Note "A": This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

(For the Calendar Year Ended December 31, 1940)

	Actual	Budget Provision	Increases Decreases
INCOME:			
Initiation fees	$ 1,875.00	$ 1,500.00	$ 375.00
Dues	28,879.29	25,000.00	3,879.29
Editorial contracts	17,050.79	16,000.00	1,050.79
Profit—Emblems and certificate frames	75.61	100.00	24.39
Profit—Reprints and books	255.89	100.00	155.89
Sales of TRANSACTIONS	967.46	500.00	1,467.46
Interest—Savings accounts and securities	1,353.71	1,300.00	53.71
GUIDE advertising	28,848.81	26,000.00	2,848.81
GUIDE sales	18,697.98	20,000.00	1,302.02
	$96,069.62	$90,500.00	$ 5,569.62
EXPENSES:			
General Society Activities:			
Meetings	$ 1,617.36	$ 2,000.00	$ 382.64
Chapter allowances	1,150.00	1,200.00	50.00
Promotion expense		500.00	500.00
Speakers Bureau	1,288.76	1,200.00	88.76
Subscriptions—HPAC	6,406.34	6,500.00	93.66
Postage	1,738.70	2,000.00	261.30
Rent and light	3,623.73	3,800.00	176.27
TRANSACTIONS	3,900.00	3,900.00	.....
Codes		500.00	500.00
Salaries—Secretary and staff	17,907.00	18,000.00	93.00
General printing	471.83	600.00	128.17
Membership certificates	188.90	500.00	311.10
President's fund	1,757.66	1,800.00	42.34
Secretary's travel	1,259.15	1,300.00	40.85
Council and chapter delegates travel	3,447.43	3,500.00	52.57
Bank charges	461.34	100.00	361.34
Multigraphing	478.93	350.00	128.93
Telephone	662.55	700.00	37.45
Telegraph	209.23	300.00	90.77
Professional services	850.00	1,000.00	150.00
General office expense	496.94	600.00	103.06
Office supplies	466.87	650.00	183.13
Furniture and fixtures	428.24	240.00	188.24
Reserve for additional compensation	2,661.19	2,800.00	138.81
Special appropriation	5,188.50	760.00	4,428.50
	$56,660.65	$54,800.00	$ 1,860.65

GUIDE EXPENSES:			
Copy sales promotion	$ 2,906.01	$ 3,000.00	$ 93.99
Express and mailing	3,440.76	4,000.00	559.24
Advertising sales promotion	1,405.07	3,000.00	1,594.93
Editorial and advertising salaries	5,591.36	5,500.00	91.36
Committee expense (chairmen's fund)	514.76	500.00	14.76
Printing and binding	13,218.90	14,000.00	781.10
Paper	3,184.29	4,000.00	815.71
Cuts and drawings	619.14	500.00	119.14
Yearbook	1,400.00	1,200.00	200.00
	$32,280.29	$35,700.00	$ 3,419.71
	$88,940.94	$90,500.00	$ 1,559.06

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS RESEARCH FUND—NEW YORK, N. Y.

(December 31, 1940)

ASSETS

CASH			
ON DEPOSIT			
Treasurer's account—Bankers Trust Co.	$2,973.77		
Secretary's account—Chase National Bank	4,634.44		
Directors' account—Forbes National Bank	401.41		
Thrift account—Bank for Savings	3,203.82	$11,213.44	
ON HAND FOR DEPOSIT			
Treasurer's account—Bankers Trust Co.		$ 1,604.53	
ON HAND			
Petty cash—Pittsburgh		16.29	$12,834.26
ACCOUNTS RECEIVABLE			
RESEARCH CONTRACTS			
Navy Department			3,561.75
PERMANENT			
Laboratory equipment			1.00
Total Research Fund assets			$16,397.01
RESEARCH ENDOWMENT FUND			
CASH ON DEPOSIT			
Bank for Savings		$ 536.83	
CASH IN CLOSED BANK			
Bank of U. S. in liquidation		167.78	$ 704.61
			$17,101.62

LIABILITIES AND FUND

DEFERRED INCOME			
Exposition income	$5,033.10		
Summer cooling	2,050.00		
Glass study	792.13		
Air distribution	250.00		
Frictional flow water	190.00		
Radiation and comfort	50.00	$ 8,365.23	
RESEARCH FUND		8,031.78	
Total Research liabilities and fund			$16,397.01

Note. "A": This balance sheet is subject to the comments contained in the letter attached to and forming a part of this report.

Note "B": The Research Fund as at December 31, 1940, had a contingent asset amounting to $4,232.77 arising from 40% of the Society's members and associates dues, payable upon collection by the latter.

RESEARCH ENDOWMENT FUND		
Principal	$ 600.00	
Unexpended income	104.61	$ 704.61
		$17,101.62

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS RESEARCH FUND—NEW YORK, N. Y.

(For the Calendar Year Ended December 31, 1940)

	Actual	Budget Provision	Increases Decreases
INCOME:			
ASHVE dues	$16,437.46	$15,000.00	$ 1,437.46
ASHVE appropriation	5,000.00	5,000.00	....
Exposition	5,000.00	5,000.00	....
Interest	55.43	60.00	4.57
Contributions:			
General	2,375.00	2,500.00	125.00
Earmarked:			
Navy contract	10,000.00	10,000.00	....
Glass	846.83	846.83	....
Radiation and comfort (Pierce)	2,017.69	2,017.69	....
Corrosion	500.00	500.00	....
Convector	400.00	400.00	....
Comfort and low humidity	3,250.00	3,250.00	....
Air distribution	750.00	750.00	....
	$46,632.41	$45,324.52	$ 1,307.89
Unbudgeted:			
Floor losses	$ 1,000.00	....	$ 1,000.00
Frictional flow water	650.00	....	650.00
Glass (additional)	407.87	....	407.87
Summer cooling	1,250.00	....	1,250.00
Window shading	2,000.00	....	2,000.00
	$ 5,307.87	....	$ 5,307.87
	$51,940.28	$45,324.52	$ 6,615.76
EXPENSES:			
Cooperative Research:			
Institutions	$11,977.60	$12,500.00	$ 522.40
U. S. Navy	10,000.00	8,612.60	1,387.40
Glass requirements	846.83	846.83	....
Radiation and comfort	2,100.25	2,017.69	82.56
Comfort low humidity	3,394.69	3,250.00	144.69
Pittsburgh Laboratory:			
Duct friction	4,613.09	5,000.00	386.91
Cooling load	2,911.53	2,700.00	211.53
Heating load	2,781.76	2,700.00	81.76
A.C. industry	3,082.61	3,000.00	82.61
Local cooling	2,105.78	2,400.00	294.22
	$43,814.14	$43,027.12	$ 787.02
Unbudgeted:			
Floor losses	$ 1,385.05	....	$ 1,385.05
Frictional flow water	650.00	....	650.00
Glass (additional)	407.87	....	407.87
Summer cooling	1,690.00	....	1,690.00
Window shading	3,107.09	....	3,107.09
Color sensation	332.52	....	332.52
	$ 7,572.53	....	$ 7,572.53
	$51,386.67	$43,027.12	$ 8,359.55

The Report of the Guide Publication Committee was given by the Chairman as follows:

Report of Guide Publication Committee

Press work on the 1941 edition of THE GUIDE is nearing completion and copies will soon be distributed.

In accordance with established policy, a considerable part of the text material has been reviewed or rewritten by authors who were selected because they were acknowledged authorities on the subjects covered. Much old material has been retained and that covering new developments has been added.

There are 46 chapters as in the previous edition but there has been some rearrangement of material. Chapter 23, Cooling, Dehumidification and Dehydration and Chapter 24, Refrigeration are new chapters containing material formerly in two old chapters plus a considerable amount of new information. The chapters have been rearranged and grouped into seven sections to make a more logical arrangement and to better correlate the Technical Data Section and the Catalog Data Section. It was necessary to revise the numbering by which some of the first chapters have been known for several years but the advantages of the change appear to be justified.

In all, 21 chapters were reviewed or rewritten this year and minor changes were made in other chapters. Chapter 1, Thermodynamics of Air and Water Mixtures, is entirely new. Since the treatment is new, it is given in greater detail than is customary in the GUIDE, for the benefit of readers who may wish to follow such a detailed discussion. A new chart, Mollier Diagram for Moist Air, is presented for use in analyzing air conditioning processes. It is printed on the sheet containing the Bulkeley Psychrometric Chart and will be found in the pocket attached to the back cover.

Other chapters which have been rewritten or revised are: Heat Transmission Coefficients and Tables, Heating Load, Cooling Load, Chimneys and Draft Calculations, Automatic Fuel Burning Equipment, Heat and Fuel Utilization, Central Systems for Comfort Air Conditioning, Spray Equipment, Fans, Air Distribution, Air Duct Design, Sound Control, Automatic Control, Air Conditioning in the Treatment of Disease, Transportation Air Conditioning, Industrial Air Conditioning, Electric Heating and Terminology.

The Catalog Data Section contains much useful information which has been supplied by the various manufacturers. The equipment has been grouped in sub-divisions for convenience in locating data.

There is an increase of 22 pages in THE GUIDE this year but thin paper has again been used so that the thickness of the book is approximately the same as for the previous edition. The comparison with the 1940 edition is as follows:

	1941	1940
Technical Data Section	808 pp.	792 pp.
Advertising Space	260 pp.	244 pp.
Total	1206 pp.	1184 pp.
Number of copies	11,000 copies	10,500 copies
Sales	$20,000.00	$18,697.98
Advertising Revenue	$29,979.25	$28,010.00

The Committee held three meetings, two in New York and one in Washington. The expenses of the Committee were kept within the budget of $500.00 provided by the Council.

The Committee is appreciative of the response of the membership to the request which it made for suggestions for THE GUIDE. Many valuable suggestions were received. All were carefully considered and as many as possible were incorporated in this edition.

The text is made up of the contributions of many individuals over a period of years. To those who have contributed to previous editions the material which makes up the greater part of the Technical Data Section, the Committee acknowl-

edges its indebtedness. Thanks is particularly due to the twenty-four men who have contributed the new material which appears in this edition and whose names are listed in the preface.

The Committee on Research, the ASHVE Research Laboratory and several individuals have been particularly helpful. They have provided much new and needed information. Technical papers presented before the Society have also been the source of useful material.

The cooperation of A. V. Hutchinson was much appreciated and I want to especially thank John James for his part in the production of THE GUIDE. He has handled much of the detail of its preparation. The fact that THE GUIDE has been completed and will be out on time is largely due to his efforts. The loyal cooperation of Mr. James was of the utmost assistance. I also want to thank the other members of the Committee who worked faithfully to make the 19th edition of THE GUIDE a success.

Respectfully submitted,

S. S. SANFORD, *Chairman*,
Guide Publication Committee.

The Report of the Council is given as follows:

Report of Council

Since the last Annual Meeting five meetings of the Council were held, at Cleveland, Chicago, Washington, D.C., Houston and Kansas City.

At the Cleveland Meeting President Giesecke announced the personnel of Council and special committees which have served during the past year. Depositories for Society funds were selected and a Certified Public Accountant was appointed to audit the Society's books.

The Council authorized a special appropriation of $5,000.00 for Research and approved the General Society Budget for the calendar year 1940 on the basis of estimated income to $90,500.00 and expenditures of a similar amount. Payment of round-trip railroad fares for Chapter Delegates to the Annual Meeting was approved and authorization given for similar payment to Council Members who attended the Annual and Semi-Annual Meetings held during the calendar year.

Meeting invitations from several chapters were considered and the Council accepted the invitation of the Kansas City Chapter for the 47th Annual Meeting and decided upon San Francisco as the meeting place for the Semi-Annual Meeting 1941 and authorized the sponsorship of a Pacific Heating and Air Conditioning Exposition.

Washington, D. C. was selected for the Semi-Annual Meeting 1940 and Houston, Tex., was approved as the meeting place for the Fall Meeting in October 1940.

The technical programs for the 1940 meetings were approved as submitted by the Meetings Committee.

On petition of the Iowa and Nebraska members, charters were granted for the Nebraska Chapter with headquarters at Omaha and the Iowa Chapter with headquarters at Des Moines, to replace the original Iowa-Nebraska Chapter. A charter was granted also for the Connecticut Chapter with headquarters at New Haven.

During the year the Membership Committee revised the booklet on *The Aims and Activities of the Society*.

The Council accepted the recommendations of a special committee on the renewal of the editorial contract for a five-year period.

The Publication Committee was requested to make a detailed study of publication policy. The suggestions of the Meetings Committee for those who prepare and present papers before the Society were approved. Renewal of the lease on headquarters office space for a three-year term was approved.

As requested by the Council the Committee on Constitution and By-Laws offered suggestions for revision of those sections governing membership requirements. Initiation fees, reinstatement fees and dues rates for 1941 were approved and the Exchange Service Plan with *ASRE* was renewed.

A request to solicit advertising by a local chapter to defray the cost of issuing the membership directory was disapproved by the Council.

A guest register was prepared and presented to the Institute of Boiler and Radiator Manufacturers for their newly established IBR Research Home.

The Council took action in accordance with requirements of the Constitution in nominating candidates for election to the Committee on Research who voted on the recommendations of the Committee on Admission and Advancement, approving the election of candidates to membership and acted on the cancellation of membership of those who were in arrears in dues. At the final meeting of the Council unfinished business was acted upon and resolutions of appreciation for President Giesecke and the retiring Council Members were adopted.

Respectfully submitted,
A. V. HUTCHINSON, *Secretary.*

The Report of the Secretary is presented as follows:

Report of Secretary

The activities of the Society for the year 1940 were equivalent to a business of nearly $150,000 of which over $50,000 represented Research. The various reports of Officers and Committees will show that a year of excellent progress has been recorded.

The headquarters office has had the responsibility of carrying on the routine membership functions, with its innumerable details of elections, changes of addresses, surveys, employment, correspondence and bookkeeping; preparing the monthly JOURNAL, TRANSACTIONS, Vol. 45 and the mechanical production of THE GUIDE, as well as the solicitation of advertising and sale of copies.

Membership applications from 272 were received and the present total membership of 3003 shows a net loss of 142 members. Several unusual factors have been involved this year, namely, difficulty of foreign members in making dues payments with resulting heavy losses, entry of members into military service who have requested temporary suspension of their membership. The offsetting factor has been the higher rate of collection of back dues and the fact that there are fewer now on the rolls in arrears than a year ago.

In a membership survey it is evident that many changes have been made by members and the address records show as many as four changes for one man, within a period of 10 months. The trend may be estimated in the number of address changes that have been recorded this year totaling 1468. To keep the address file up to date is one of the most difficult assignments and at the present time many members are now listed under "mail returned" classification. Sometimes after several weeks or months a member writes from another city and complains about the non-receipt of his publications. To avoid this, full cooperation in sending address changes promptly is essential.

Many members are on national defense projects and it is interesting to note that the average age of our members makes 37 per cent of those in the U. S. eligible under the draft law.

For the Society's three meetings during the year, the headquarters office has cooperated with the Local Committees on Arrangements in publicizing the meetings to members and close cooperation has been carried on with the Meetings, Publication, Finance, Guide, Membership, Research and numerous other special and technical Committees. The Secretary has had the opportunity of visiting 17 of the Society's 31 chapters during the year and all 31 chapters have received an official visit from President Giesecke, or Vice-President Fleisher and Eastwood.

The Speaker's Bureau has been called upon by 27 chapters for the assignment of 44 speakers during the year and because of limited funds it was necessary during the last quarter to enlist the services of volunteers. A number of the Council and committee members offered their services without expense to the chapters or the Society. Because of geographical location it was not possible to provide all chapters with a list of speakers. For the better administration of the Speakers Bureau it would be advisable for the Council to amend some of the rules for the coming year.

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Although the work involved in handling the administrative duties of the Society has increased considerably in volume during the past year, it has been possible to meet all demands without increasing the staff as everyone has given full cooperation.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

The second technical session convened at 10:00 a.m. on Tuesday, January 28 with First Vice-President Fleisher presiding. The first item presented was the Report of the Committee on Research which was given by its Chairman, A. E. Stacey, Jr. as follows:

Committee on Research

During 1940 the Society's research activities have resulted in the completion of nine laboratory projects and the initiation of eight new investigations at the ASHVE Research Laboratory and at cooperative institutions. This work has produced valuable scientific papers and reports contributing needed information on heating, ventilating and air conditioning to the engineering profession and to industry. Earlier in the year the available facilities of the research organization were offered by the President of the Society to the federal government for service in connection with the national defense program.

Completed laboratory investigations have dealt with the effect of radiation on the sensation of warmth of an occupant in an enclosed space, and reactions of individuals to environment having low relative humidities. Additional studies have also developed data for establishing heat gain through windows with various types of appurtenances; frictional flow of water in large size pipes; and comparative performance of air filters utilizing various methods of test. Among the new investigations which have been outlined and initiated are those dealing with noise transmission in ducts; heat losses from basements; air flow in duct transitions; and studies to establish basic design factors in radiant heating and cooling applications.

The research program was conducted through 23 Technical Advisory Committees, each responsible for reviewing, analyzing and improving through research the fundamental data in its particular field or scope. A total expenditure of $45,000 was spent by the Society on the various projects. Effort was made this year to clearly define the scope of the activities of each Technical Advisory Committee in order that the whole research program could be correlated to produce basic fundamental data.

The reports which follow give detailed information covering the committee programs developed for investigation either at the Research Laboratory or in cooperative institutions.

Reports of Technical Advisory Committees

SENSATIONS OF COMFORT, *1*—Thomas Chester, *Chairman;* C. R. Bellamy, E. P. Heckel, W. S. Kilpatrick, Dr. W. J. McConnell, A. B. Newton, B. F. Raber, C. Tasker.

This Committee work has progressed along the lines set forth in the annual report for 1939. Eleven papers dealing with indoor atmospheric conditions conducive to summer comfort in air conditioning enclosures have been reviewed. These

papers cover information ascertained by the ASHVE Research Laboratory and in field tests at San Antonio, Washington, D. C., New York City, Minneapolis and Toronto.

A map has been prepared which shows the United States divided into four zones on the basis of July mean-maximum temperatures with corresponding recommended indoor effective temperatures of 70, 71, 72 and 73 deg in the order stated for the four zones from north to south. This map will be published subsequently along with a digest of information on summer comfort contained in the eleven papers previously mentioned.

Investigative work has been conducted during 1940 at the Research Laboratory to determine the effects of low relative humidities on comfort. A paper covering this work will be presented at the 1941 Annual Meeting.

PHYSIOLOGICAL REACTIONS, *2*—C.-E. A. Winslow, *Chairman;* Dr. T. Bedford, Dr. E. F. DuBois, Dr. M. B. Ferderber, E. P. Heckel, John Howatt, Dr. R. W. Keeton, C. S. Leopold, André Missenard, Dr. R. R. Sayers, Charles Sheard, C. Tasker.

This Committee is a continuing group having the responsibility of presenting at intervals summaries of current physiological researches and their bearing on ventilation practice. A comprehensive report was presented at the 1939 Annual Meeting and it has been the feeling of this Committee that new material bearing on this subject was insufficient to warrant a report this year. It is anticipated that by 1941 a new summary of advances in this field of knowledge will be available for presentation.

Cooperative studies conducted under the auspices of this Committee have been continued at the Medical School of the University of Illinois in Chicago dealing particularly with the important factor of blood volume as related to circulatory changes in the human body. A preliminary report covering these results will be given at the 1941 Annual Meeting and a final report should be available in the spring.

TREATMENT OF DISEASE, *3*—Dr. M. B. Ferderber, *Chairman;* Dr. C. J. Barone, Dr. A. R. Behnke, Dr. B. Z. Cashman, W. L. Fleisher, Dr. T. L. Hazlett, C. S. Leopold, Dr. C. D. Selby, Dr. A. W. Sherrill, Dr. H. F. Smyth, Dr. C. S. Stephenson, Dr. B. L. Vosburgh.

The application of air conditioning to operating rooms and recovery wards has been the principal objective of the Committee. One investigation has dealt primarily with the bacteriological content of air introduced to an operating room as the result of cleaning with various types of available equipment. A report covering this work has been released indicating that the average bacterial count in an operating room was decreased with available air cleaning equipment in operation. Cooperative efforts have been initiated with interested groups to study the explosive hazards of combustible gases and vapors commonly used for anesthetic purposes.

AIR CONDITIONING IN INDUSTRY, *4*—W. L. Fleisher, *Chairman;* Philip Drinker, Dr. Leonard Greenburg, H. P. Greenwald, A. M. Kinney, J. W. Kreuttner, L. L. Lewis, H. B. Matzen, Dr. W. J. McConnell, Dr. C. P. McCord, P. A. McKittrick, Dr. R. R. Sayers, Charles Sheard, C. Tasker, R. M. Watt, Jr.

Work conducted at the Research Laboratory this year under the direction of this Committee has dealt primarily with determining differences in physiological reactions of persons in the same atmospheric environment at a low moisture content to check results previously obtained at high relative humidities. A few of the preliminary findings of this study will be reported in a paper presented at the 1941 Annual Meeting and a more comprehensive report of this work will be included in a paper to be prepared for subsequent publication.

Additional observations were made this year to note changes in the leucocyte count. Two other points of investigation are being considered: (1) to obtain data on the relative physiological changes of men and women in exposure to hot environments; and (2) to obtain physiological reactions of persons at different rates of activity to atmospheric environment.

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AIR POLLUTION AND PURIFICATION, *5*—C.-E. A. Winslow, *General Chairman.*

Air Pollution: H. J. Rose, *Chairman;* Philip Drinker, H. B. Meller, J. S. Owens, Sol Pincus, R. J. Tenkonohy, E. C. Webb, E. H. Whitlock.

In accordance with the policy determined at the organization meeting of this Committee in December, 1939, the work of this group during the past year has been limited principally to a study of the literature relating to sources of outdoor air pollution, methods of controlling such pollution at the source, and a review of urban air pollution surveys. The purpose of this survey is to provide a basis for orientation in planning future work of the Committee.

Removal Atmospheric Impurities: Dr. Leonard Greenburg, *Chairman;* R. S. Dill, Theodore Hatch, L. R. Koller, F. H. Munkelt, G. W. Penney, Dr. E. B. Phelps, F. B. Rowley, C. Tasker, W. O. Vedder, Jack Waggoner, R. P. Warren, W. F. Wells, Dr. Renée Wiener.

The work of the Committee is divided into three sub-groups formulated to deal with special aspects of the problem—namely, mechanical, bactericidal and chemical.

The *mechanical* committee has endeavored to obtain factual data for some of the various problems which will have to be dealt with in the future by this Committee. Samples of various types of dust have been collected as they are found in various industrial operations at the point of production, at the effluent from the collector and of the material itself collected by the removal device. These samples have been submitted to careful microscopic particle size determinations and separated into various classifications as follows: (1) under 20 microns, (2) under 10 microns, and (3) under one micron. The findings of this portion of Committee activities have not yet been completely analyzed, but it is planned to extend this work until sufficient data have been obtained to represent a satisfactory picture of the usual type of industrial dust producing processes and dust filtration devices.

The *bactericidal* sub-committee has been active during the past year in the development of literature and the critical examination of experimental data evolved by various members of this group. An extensive report is now being prepared which will be available for publication in January, 1941.

Activity of the *chemical* sub-committee has been limited due to the inability of the personnel to function in view of concentrated work surrounding the defense program of the government.

RADIATION AND COMFORT, *6*—J. C. Fitts, *Chairman;* A. H. Barker, L. M. K. Boelter, R. E. Daly, E. R. Gurney, L. N. Hunter, A. P. Kratz, C. S. Leopold, D. W. Nelson, W. J. Olvany, G. W. Penney, W. R. Rhoton, C.-E. A. Winslow.

This Committee was appointed in 1938 to consider all problems relating to radiation as it affects comfort in winter and summer. A comprehensive outline of basic researches was prepared and it was decided to concentrate first on the study to determine the primary sense reactions of relative comfort derived from rooms with normal window arrangements from conducted heat and from heat supplied by both convection and direct radiation.

During the summer of 1939 the necessary building was erected at the ASHVE Research Laboratory in Pittsburgh and throughout the winter of 1939 and 1940 tests were conducted, the results of which will be submitted in a paper to be presented at the Annual Meeting.

It is planned to continue further work on this project at the Research Laboratory during 1941. In addition, the Committee will be responsible for directing a cooperative project at the University of California dealing with radiant heating and cooling.

INSTRUMENTS, *7*—D. W. Nelson, *Chairman;* F. R. Bichowsky, L. M. K. Boelter, R. S. Dill, M. K. Fahnestock, A. P. Gagge, J. A. Goff, F. W. Reichelderfer, G. L. Tuve, C. P. Yaglou.

The development of standards for measuring thermal environment, and more particularly, a uniform appraisal of mean radiant temperature has been the scope of activity for this Committee during the current year. A coordinated investigation of the various factors dealing with thermal environment and physiological reactions is being made at several of the cooperative institutions and the Research Laboratory.

Tentative standards have been approved for blackening globe thermometers and experimental work has been conducted to show the influences of type and location of the instrument for measuring mean radiant temperature. A low velocity hot wire anemometer for measuring air movement at the location of a globe thermometer has been developed although further analysis and standardization is contemplated.

The problem of uniform air measurement in ducts, at various outlets and inlets and in free spaces is being studied in several laboratories doing cooperative research on air distribution. Out of this experience it is expected that the range and method of application for various instruments will finally be determined. No one instrument can be expected to accurately measure all of the varying conditions encountered in air flow problems.

WEATHER DESIGN CONDITIONS, *8*—T. H. Urdahl, *Chairman;* J. C. Albright, P. D. Close, John Everetts, Jr., C. M. Humphreys, J. B. Kincer, O. A. Kinzer, J. W. O'Neill, F. W. Reichelderfer.

During the past year the Committee's attention and its efforts have been focused upon accomplishing the objectives set forth in its program adopted in 1938. The results of this work of the Committee during 1938 provided valuable additions and corrections to be applied on existing design data, and as reported at that time, established through its work a basis of operation upon which to continue. The objectives as previously reported are repeated herewith as follows:

1. To establish uniform methods of obtaining and analyzing weather data upon which design conditions for any given locality may be based.

2. To quantitatively analyze existing weather data including dry- and wet-bulb temperature, wind velocity, and sunshine, in a manner and over a sufficient period of years to show range and duration by months and seasons following the procedure already employed in obtaining some of the summer data.

3. To supplement the work being done by the U. S. Weather Bureau through obtaining those data which they do not have available. The chief deficiency is wet-bulb data which are only available for approximately four years in most localities. This supplemental work to consist of the taking of wet-bulb and dry-bulb temperatures at a central point in each locality at the street level and at the average roof level for comparison with local temperature differences between business and suburban sections and the influence of solar heat storage in massed pavements, buildings, etc.

4. To make available the information obtained in the foregoing to each locality through assembly, analysis, and distribution from one central point until such time as such work may be done by the Government as a part of its weather reporting program.

5. To recommend to the U. S. Government desired additions to the changes in the data being obtained through the Weather Bureau stations, and to support the required legislative effort as may be necessary to expand the activity of this bureau to include the reporting and dissemination of the data required in order to form a proper and adequate basis for design of heating and air conditioning systems.

Item 1 will be ultimately accomplished upon completion of analytical work under way. The balance of the items are the subject of present cooperative effort with the U. S. Weather Bureau now under the Department of Commerce, whose expanded facilities and appreciation of the Committee's needs are making it possible to obtain basic data which will be of greater accuracy for local application than that upon which engineers have had to depend in the past. Details of this program are enumerated herewith.

A. The U. S. Weather Bureau has available approximately 14,000,000 hourly airway observations of temperature and dewpoint which are being analyzed to produce tables of design load data heretofore unavailable. Through correspondence with Committeee personnel, representatives of industry and individuals in various departments of the government, six tables of various arrangements of hourly readings of dry-bulb, wet-bulb and dewpoint temperature were agreed upon as being of greatest value to the designing engineer. These tables with some slight variations necessary to facilitate compilation will be prepared under the direction of the Weather Bureau in cooperation with a government project located at New Orleans which will employ approximately 200 individuals on the work. In setting up this program, preliminary to the actual compilation, some 29,000 forms are being printed and from

1 to 1½ years will be required to complete the work. These tables will include data for 115 stations throughout the United States and the order in which they will be prepared will likely be based upon the population served by the respective stations.

B. In order to determine local temperature differences mentioned in Item 3 of the basic program, cooperative work is in the process of being initiated in Washington, D. C., through the establishment of several recording stations at representative points taking into consideration geological characteristics of the urban and suburban sections of the city. A comparison of the simultaneous readings obtained with the official Weather Bureau reading for the locality will establish a percentage of variation for sections. The conduct of this work in Washington will enable the development of a procedure which may be applied to other localities in order to obtain these data for each representative city. In this work, the cooperation of local utilities, colleges and universities and especially local Society Chapters is being earnestly sought to provide the personnel required for making the observations.

C. In the recording and abstraction of data for the summary of meteorological data published every 10 years, the Committee has cooperated with the U. S. Weather Bureau along lines of value to the profession and industries concerned with heating, ventilating and air conditioning.

RADIATION WITH GRAVITY AIR CIRCULATION, 9—M. K. Fahnestock, *Chairman;* R. E. Daly, R. S. Dill, A. G. Dixon, H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, T. A. Novotney, W. A. Rowe.

Under this Committee, work during 1940 has been a continuation of a program in progress for a number of years as a cooperative project with the University of Illinois. The work may be divided into three parts—*first,* studies for sponsoring and improving methods of testing and rating radiators and convectors, *second,* a study of the effect of the temperature of a cold room upon the heat output of radiators and convectors, and, *third,* study of the effect of steam temperature upon the heat output of convectors. The program for standardization of test was initiated more than a year ago and has been conducted in cooperation with seven interested laboratories and the Bureau of Standards. Three convectors, two with cast-iron heating units and one with a non-ferrous heating unit, and two direct cast-iron radiators are being tested in each laboratory. The results of the tests in each laboratory are to be studied and compared by the staff at the University of Illinois.

In laboratories which are equipped with test rooms surrounded by cold space the temperature of the air in this space is not always uniform. In order to obtain reasonable temperatures in exposed test rooms lower temperatures are maintained in the cold space when large capacity units are tested than when smaller capacity units are tested. To determine if the temperature of the air in the cold space has any appreciable effect upon the heat output, two types of radiators and convectors are being tested in a cold wall experimental room with a varying range of temperatures in the cold space.

Studies to determine the effect of steam temperature upon the heat output of convectors involves testing two types of convectors using varying steam temperatures and inlet air temperatures to ascertain whether there is a difference in the heat output of such units.

Tests will be made in the immediate future to determine whether or not the number of exposed walls in a cold wall test room has any appreciable effect upon the heat output per unit of surface of a direct radiator or per unit of length of convector.

HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION, *10*—W. E. Heibel, *Chairman;* William Goodman, H. F. Hutzel, Ferdinand Jehle, S. F. Nicoll, R. H. Norris, L. P. Saunders, R. J. Tenkonohy, G. L. Tuve, C. F. Wood.

Considerable experimental data collected at Case School of Applied Science through a cooperative agreement with the Society, dealing with several methods of calculating dehumidifying coils applied to the same design conditions, have been analyzed by the Committee this year. A code for the testing and rating of finned tube coils has been discussed by the Committee personnel with the prospect of making definite recommendations to the code committee of the Society dealing with this subject at an early date.

COOLING LOAD IN SUMMER AIR CONDITIONING, *11*—C. M. Ashley, *Chairman;* John Everetts, Jr., M. G. Kershaw, A. E. Knapp, C. S. Leopold, L. S. Morse, R. M. Stikeleather, J. H. Walker.

During the summer of 1939 a cubicle was built at the Research Laboratory for the purpose of studying the effect of sun radiation on heat gain through building walls and roofs. The details of the wall and roof construction are given in the 1939 annual report. During the summer of 1939 many data were accumulated but, unfortunately, some of the wall data were found to be incomplete due to a shift in calibration of the Nicholls' heat flow meters used to determine the rate of heat flow through the walls. A good share of the data on the roof were satisfactory, however, and were reported in a paper read before the 1940 Summer Meeting of the Society.

During the summer of 1940 additional data were obtained on the walls after having aged and recalibrated the Nicholls' heat flow meters. These data will be the subject of a forthcoming paper.

It now appears that the magnitude of sun effect and lag on roofs and walls has been established; also the relative effect of different wall facings. The present cubicle provides information on only three types of wall construction. Two synthetic methods of predicting the results for other wall constructions have been tried; one a purely mathematical method, the other a hydraulic analogy for heat flow. These are only partially satisfactory, since they depend upon the outside surface temperature as a starting point.

The Committee is also considering the advisability of further experimental work on other wall constructions in order to complete the information on this subject. This would require the construction of another cubicle. Proposals are also under consideration to study the heat storage effect of interior walls due to fluctuation of room temperature, infiltration through swinging and open doors, and heat from appliances.

SOLID FUELS, *12*—R. A. Sherman, *Chairman;* C. E. Bronson, W. A. Danielson, R. S. Dill, H. N. Eavenson, A. C. Fieldner, L. N. Hunter, A. J. Johnson, R. E. Kerr, H. K. Kugel, F. L. Meyer, P. Nicholls, V. F. Parry, H. J. Rose, L. E. Seeley, C. E. Shaffer, C. Tasker, E. C. Webb.

Marked by a thorough discussion of the many problems of the utilization of solid fuels in heating, the one meeting of this committee, held during the Annual Meeting in Cleveland, laid plans for a thorough review of the status and needs for codes relating to solid fuels. Sub-committees charged with the responsibility of reviewing the codes of the boiler industry, the warm air furnace industry, and of the stoker industry have been actively working on their assignments.

Plans have been formulated to develop a fundamental chimney research program with one phase of the work to be done at a cooperative institution and to encourage collection of data on chimneys at other laboratories where research on heating is now being conducted. It is hoped that the information can be coordinated into a useful collection of chimney data.

The project on the combustion of sub-bituminous coals on residential stokers at the Golden, Colorado Station of the U. S. Bureau of Mines is going forward and is being broadened in scope but no publication of results has yet been made.

SUMMER AIR CONDITIONING FOR RESIDENCES, *13*—M. K. Fahnestock, *Chairman;* C. F. Boester, E. A. Brandt, John Everetts, Jr., H. F. Hutzel, E. D. Milener, K. W. Miller, F. G. Sedgwick, J. H. Walker.

All of the active work of this Committee during 1940 was conducted at the warm air heating Research Residence at the University of Illinois. The primary object of the tests conducted in the past summer was to secure operating and performance data on the insulated residence in order to compare these data with similar data obtained with the same residence uninsulated. Dehumidifying equipment was installed with a dry air cooler and resaturation cooler for maintaining indoor conditions from 78 to 80 F dry-bulb and from 45 to 50 per cent relative humidity. These conditions are comparable to those maintained in 1934 when mechanical cooling was used in the uninsulated residence. The primary series of tests was conducted using circulated outdoor air at night when feasible and additional tests were secured keeping

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the house closed both during the day and night. A paper covering the results of these tests will be presented at the Annual Meeting.

AIR DISTRIBUTION AND AIR FRICTION, *14*—J. H. Van Alsburg, *Chairman;* S. H. Downs, A. E. Hershey, R. D. Madison, L. G. Miller, D. W. Nelson, C. H. Randolph, D. J. Stewart, M. C. Stuart, Ernest Szekely, R J. Tenkonohy, G. L. Tuve.

The work of this Committee has been in progress for a period of four and a half years with investigations being conducted at the Research Laboratory and in five cooperative institutions. Air flow in ducts has been the subject of investigation at the Research Laboratory for a period of four years which has resulted in two papers and a third will be available for presentation at the 1941 Summer Meeting of the Society. All of this work has dealt primarily with the friction of air in straight ducts. Research on elbows of various radii is being conducted at Lehigh University and a paper covering the first portion of this work will be available for presentation at the next Summer Meeting of the Society. The third project in this field is being investigated at Michigan State College to ascertain the frictional flow characteristics in various types of duct transitions.

In the field of air distribution it was found necessary to first develop a method of testing and correlating instruments on a uniform basis. Some of the first work on the development of instrument standardization was done at Case School of Applied Science which resulted in a paper being presented at the 1940 Fall Meeting of the Society. Additional work at this institution was done this year on the characteristics of supply openings and the air streams delivered into unconfined spaces.

At the University of Wisconsin investigative work has been done to determine the effect of stack heads and approaches on the expansion of an air stream as it leaves a supply opening. This program has resulted in a paper being presented at the 1940 Annual Meeting and another report is now in preparation for publication at an early date.

Another project on air distribution involves the effect of inlet and exhaust opening location on the motion and distribution of air within a room which is being conducted at the University of Illinois. The primary objective of this program is to determine the location of air inlet and outlet to a room which will give the most uniform temperature throughout a room and still provide no objectionable air motion. A preliminary paper covering this work was presented at the 1940 Fall Meeting and it is expected that additional papers covering this work will be available for presentation in the near future.

HEAT REQUIREMENTS OF BUILDINGS, *15*—P. D. Close, *Chairman;* E. K. Campbell, J. F. S. Collins, Jr., E. F. Dawson, W. H. Driscoll, H. M. Hart, E. C. Lloyd, H. H. Mather, C. H. Pesterfield, F. B. Rowley, R. K. Thulman, J. H. Walker.

Through the support of earmarked funds a cubicle 15 x 15 x 8 ft high was built on the premises of the Bureau of Mines in Pittsburgh and located in the ground to simulate an actual basement. It is expected that such a structure will have the advantage of eliminating interference from occupants and will permit the maintenance of constant conditions. Thermocouples have been placed at various locations which will be recorded by the staff at the Research Laboratory. Insulation has been placed on the outside of two walls of the cubicle and although no insulation has been placed in the floor it is expected that this can be added later for future tests. The structure has an insulated ceiling and is electrically heated with the temperature thermostatically controlled. The testing program is planned to study the heat flow characteristics through basement walls, ground temperatures, heat loss coefficients and possible summer condensation.

In addition the Committee has under consideration a proposal for the study of heat transmission through wet wall conditions and a mathematical study of the heating load for intermittently heated buildings.

AIR CONDITIONING REQUIREMENTS OF GLASS, *16*—M. L. Carr, *Chairman;* C. M. Ashley, L. T. Avery, F. L. Bishop, W. A. Danielson, H. C. Dickinson, J. D. Edwards, J. E. Frazier, S. O. Hall, E. H. Hobbie, C. L. Kribs, Jr., Axel Marin, R. A. Miller, F. W. Parkinson, J. H. Plummer, W. C. Randall, L. T. Sherwood, J. T. Staples, G. B. Watkins, F. C. Weinert.

Two meetings of this Committee were held during the current year and plans were formulated for initiating a comprehensive investigation at the Research Laboratory on the value of sun arresting appurtenances. This program was planned to yield information most urgently needed and to constitute a general survey of the subject. A paper covering tests which could be completed in the limited time available will be presented at the Annual Meeting.

The solar heat gain measurements were made with the turntable-mounted twin-cubicles built for use in a previous study of solar heat gain through glass blocks. In general, each test consisted of measuring the simultaneous heat gain through two different kinds of window shading. Devices tested included an awning, an outside venetian blind, an inside venetian blind, an outside shading screen, a shading screen outside and a venetian blind inside and a half-drawn roller shade. In addition, simultaneous tests were made upon single-glazed and double-glazed glass, heat absorbing glass and upon wood and steel sash.

INSULATION, *17*—R. T. Miller, *Chairman;* E. A. Allcut, R. E. Backstrom, Wharton Clay, R. E. Daly, W. A. Danielson, H. C. Dickinson, J. D. Edwards, W. V. Hukill, E. C. Lloyd, Paul McDermott, E. W. McMullen, W. T. Miller, E. R. Queer, F. B. Rowley, W. S. Steele, J. H. Waggoner, G. B. Wilkes.

The principal activities of this Committee during the year 1940 dealt with the review of a memorandum on insulation for submittal to the Guide Publication Committee covering reflective materials. Consideration has been given to the preparation of a coordinated program of fundamental research which would have value to the entire industry. As a new code for testing insulating materials is now completed it is hoped that a program may be developed for retesting all insulating and building materials on a uniform basis for future inclusion in the GUIDE tables.

SOUND CONTROL, *18*—J. S. Parkinson, *Chairman;* E. L. Anderson, C. M. Ashley, A. L. Kimball, V. O. Knudsen, R. D. Madison, R. F. Norris, C. H. Randolph, A. E. Stacey, Jr., G. T. Stanton, A. G. Sutcliffe, R. M. Watt, Jr.

Research work on the nature of noise in duct systems was initiated this year through a cooperative agreement with Rensselaer Polytechnic Institute. The Committee prepared an outline of the work to be done and cooperated in selection of equipment and test procedures. At the present time the work is under way, although no formal report has yet been issued.

A meeting of the Committee was held in Washington in June of this year where a revision of the table of acceptable noise levels which appears in THE GUIDE was prepared. This revision is based on more recent surveys of typical noise levels which have been published within the last two years. Methods of rating apparatus noise were discussed and an agreement was reached in principle as to the most desirable procedure. Several members of the Committee undertook to investigate further certain aspects of this problem.

COOLING TOWERS, EVAPORATIVE CONDENSERS AND SPRAY PONDS, *19*—B. M. Woods, *Chairman;* J. C. Albright, W. W. Cockins, S. C. Coey, E. H. Hyde, E. H. Kendall, S. R. Lewis, J. Lichtenstein, H. B. Nottage, J. F. Park, E. T. Selig, Jr., E. H. Taze.

The work on cooling towers and spray ponds has progressed along several lines since the previous report of this Committee. A complete set of tests has been conducted under a cooperative agreement with the University of California on a spray pond and the results are being evaluated for publication at an early date. Also partial evaluation of the test results obtained on the atmospheric tower has been accomplished. Further tentative computations on the cooling of falling drops have been completed. A study of experimental results on cooling ponds available has been initiated. A summary of the latest techniques of cooling tower packing which involved a comparison of the behavior of approximately 10 different types of European and American towers has been prepared under the direction of the Committee for use in the 1941 edition of THE GUIDE. Developed during the past summer has been a technique for the testing of single elements of cooling tower packing. A hot wire anemometer which will record fluctuating flow and a fog-measuring device have been completed during the past year. An accurate dew-point recorder and a pressure plate type of fluctuating air flow measuring instrument have been designed.

An outline of work for tests on an evaporative condenser has been formulated which will begin as soon as the necessary equipment has been obtained to initiate the project.

PYSCHROMETRY, *20*—F. R. Bichowsky, *Chairman;* D. B. Brooks, W. H. Carrier, H. C. Dickinson, A. W. Gauger, J. A. Goff, William Goodman, A. M. Greene, Jr., L. P. Harrison, F. G. Keyes, A. P. Kratz, D. M. Little, Axel Marin, D. W. Nelson, W. M. Sawdon, F. O. Urban.

The major activity of this Committee is an exact determination of the thermodynamic properties of air-water vapor mixtures. Through a cooperative agreement with the University of Pennsylvania suitable apparatus has been designed and built for measuring the departures from Dalton's Law of air-water vapor mixtures. The apparatus is operating successfully at the present time and preliminary results have been obtained. These data are now being extended, checked and analyzed.

CORROSION, *21*—A. R. Mumford, *Chairman;* H. E. Adams, J. F. Barkley, L. F. Collins, W. H. Driscoll, T. J. Finnegan, W. Z. Friend, R. R. Seeber, E. T. Selig, Jr., F. N. Speller, J. H. Walker.

The cooperative work at Michigan College of Mining and Technology, which was interrupted during 1939 by a destructive fire, has been resumed in a new laboratory. This work is expected to furnish information on the quantities of non-condensible gases which are dissolved in the condensate formed in a heating system operated at several different rates of condensation. Some studies have been made of the operation of storage type water heaters by means of steam controlled at the condensate outlet rather than at the steam inlet. This type of control offers possibilities in the reduction of corrosion resulting from non-condensible gases. A report of a test of the corrosiveness of various materials of contruction of air conditioning equipment was discussed at a committee meeting in Washington in June of this year. It has been planned to present this report in a more extensive form in a subsequent publication, possibly with the inclusion of additional experimental data.

FLOW OF FLUIDS THROUGH PIPES AND FITTINGS, *22*—S. R. Lewis, *Chairman;* L. A. Cherry, G. C. Davis, T. M. Dugan, Earle W. Gray, R. T. Kern, H. A. Lockhart, Axel Marin, R. F. Taylor, E. L. Weber.

The activities of this Committee have been confined to a cooperative research project being investigated at the Agricultural and Mechanical College of Texas. The Committee collected a special fund to cover the cost of erecting a large section of 6-in. pipe, with valves, fittings, and sections of both welded and screwed methods of assembly.

This test was made because of the rapidly increasing use of large size piping in forced circulation hot water heating. Very little is known as to the effect of turbulence following obstructions such as by tees, and following changes in direction as at elbows, and whether the same relative frictional value due to any specific resistance in a small pipe conveying a liquid applies to a large pipe.

The influence of incrustation which may occur over a period of years on the inside of piping used for carrying recirculated hot water has had little authoritative study. The effect on ultimate resistance to flow of the oxidation which may occur when a water pipe alternately is wetted and dried also needs elucidation, since there are very wide variations in allowances by different authorities.

The tests for friction in 6-in. piping have been completed and will be presented at the Annual Meeting. They indicate that minor corrections not of serious consequence will be required in the GUIDE tables. The tests on effect of long time use under service conditions are now under investigation.

CONTRIBUTORS TO RESEARCH

Contributions in support of the research work of the Committee on Research are hereby acknowledged:

Financial Contributions

Aerofin Corp.; American Air Filter Co., Inc.; *American Gas Association;* American Marsh Pumps, Inc.; American Radiator & Standard Sanitary Corp.; Anemostat

Corporation of America; Barber-Colman Co.; Bell & Gossett Co.; Buffalo Forge Co.; A. M. Byers Co.; Carrier Corp.; Chicago Pump Co.; Crane Co.; Dertoit Lubricator Co.; Frick Company, Inc.; G & O Mfg. Co.; Grinnell Co.; *Heating, Piping & Air Conditioning Contractors National Association;* Hoffman Specialty Co.; Illinois Testing Laboratories, Inc.; Independent Register Co.; Ingersoll Steel & Disc: Div. of Borg-Warner Corp.; *Insulation Board Institute;* Johnson Service Co.; Kewanee Boiler Corp.; Kinetic Chemicals, Inc.; Libbey-Owens-Ford Glass Co.; *Metal Window Institute;* Minneapolis-Honeywell Regulator Co.; Narowetz Heating & Ventilating Co.; Nash Engineering Co.; *National Door Manufacturers Association, Inc.;* National Tube Co.; Owens-Illinois Glass Co.; Pittsburgh Corning Corp.; Pittsburgh Plate Glass Co.; Taylor Instrument Cos.; Waterloo Register Co.; Weil McLain Co.; Westinghouse Electric & Mfg. Co.; York Ice Machinery Corp.

Donors of Equipment or Material

A-V-B Manufacturing Co.; Aluminum Venetian Blind Co.; American Air Filter Co.; American Radiator & Standard Sanitary Corp.; Barber-Colman Co.; Bryant Heater Co.; Carrier Corp.; Chicago Venetian Blind Co.; Crane Co.; Detroit Lubricator Co.; Detroit Steel Products; Ellison Gage Co.; Julien P. Friez & Sons; Hart & Cooley Manufacturing Co.; Ingersoll Steel & Disc: Div. of Borg-Warner Corp.; *Insulation Board Institute;* Johns-Manville; Libbey-Owens-Ford Glass Co.; Minneapolis-Honeywell Regulator Co.; Modine Manufacturing Co.; Mueller Furnace Co.; National Radiator Co.; National Tube Co.; Owens-Corning Fiberglas Corp.; Pittsburgh Plate Glass Co.; F. C. Russell Co.; The Trane Co.; Tuttle & Bailey.

INSTITUTIONS COOPERATING WITH THE COMMITTEE ON RESEARCH

Agricultural & Mechanical College of Texas: Frictional Flow of Water in Pipes; Heating Requirements of Buildings. *University of California:* Performance of Cooling Towers; Radiant Heating and Cooling. *Case School of Applied Science:* Air Distribution in Air Conditioned Spaces. *Georgia School of Technology:* Investigation of Attic Fan Applications.

University of Illinois, Engineering: Radiator and Convector Studies; Residence Summer Air Conditioning; Air Distribution Outlets. *University of Illinois, Medical School:* Human Responses to Physiological Reactions. *Lehigh University:* Frictional Resistance to the Flow of Air Through Elbows. *Michigan College of Mining & Technology:* Corrosion in Steam Heating Systems.

Michigan State College: Air Flow in Duct Transitions. *University of Minnesota:* Methods of Rating Air Cleaning Devices. *University of Pennsylvania:* Measuring Departures from Dalton's Law of Air-Water Vapor Mixtures. *Rensselaer Polytechnic Institute:* Sound Transmission in Ducts. *University of Wisconsin:* Effect of Entering Air Temperature and Velocity on Distribution of Air in Enclosed Spaces.

RESEARCH PAPERS—1940

Society Laboratory

Seasonal Variation in Reactions to Hot Atmospheres, by F. C. Houghten, A. A. Rosenberg and M. B. Ferderber (ASHVE TRANSACTIONS, Vol. 46, 1940, No. 1153).

Analysis of the Factors Affecting Duct Friction, by J. B. Schmieler, F. C. Houghten and H. T. Olson (ASHVE TRANSACTIONS, Vol. 46, 1940, No. 1155).

Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, H. T. Olson and Burt Gunst (ASHVE TRANSACTIONS, Vol. 46, 1940, No. 1147).

Reactions of 745 Clerks to Summer Air Conditioning, by Dr. W. J. McConnell and M. Spiegelman (ASHVE TRANSACTIONS, Vol. 46, 1940, No. 1161).

Summer Cooling Load as Affected by Heat Gain through Dry, Sprinkled and Water Covered Roofs, by F. C. Houghten, H. T. Olson and Carl Gutberlet (ASHVE TRANACTIONS, Vol. 46, 1940, No. 1158).

Sensation of Warmth as Affected by the Color of the Environment, by F. C. Houghten, H. T. Olson and John Suciu, Jr. (November 1940 JOURNAL).

Comfort Requirements for Low Humidity Air Conditioning, by F. C. Houghten, H. T. Olson and S. B. Gunst (see p. 139).

Heat Gain Through Western Windows With and Without Shading, by F. C. Houghten and David Shore (see p. 251).

Radiation as a Factor in the Sensation of Warmth, by F. C. Houghten, S. B. Gunst and J. Suciu, Jr. (see p. 93).

Cooperative Institutions

Effect of Lint on Air Filter Performance, by F. B. Rowley and R. C. Jordan (ASHVE TRANSACTIONS, Vol. 46, 1940, No. 1145).

The Performance of Stack Heads, by D. W. Nelson (ASHVE TRANSACTIONS, Vol. 46, 1940, No. 1156).

The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion—The Role of Posture, by R. W. Keeton, F. K. Hick, Nathaniel Glickman and M. M. Montgomery (ASHVE TRANSACTIONS, Vol. 46, 1940, No. 1151).

Dynamic and Thermal Behavior of Water Drops in Evaporative Cooling Processes, by H. B. Nottage and L. M. K. Boelter (ASHVE TRANSACTIONS, Vol. 46, 1940, No. 1146).

Fever Therapy Locally Induced by Conditioned Air, by Dr. M. B. Ferderber, F. C. Houghten and Carl Gutberlet (ASHVE TRANSACTIONS, Vol. 46, 1940, No. 1162).

Effect of Room Dimensions on the Performance of Direct Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock, E. L. Broderick and S. Sachs (ASHVE TRANSACTIONS, Vol. 46, 1940, No. 1164).

Air Flow Measurements at Intake and Discharge Openings and Grilles, by G. L. Tuve and D. K. Wright, Jr. (ASHVE TRANSACTIONS, Vol. 46, 1940, No. 1163).

Development of Instruments for the Study of Air Distribution in Rooms, by A. P. Kratz, A. E. Hershey and R. B. Engdahl (ASHVE TRANSACTIONS, Vol. 46, 1940. No. 1166).

Summer Cooling in Research Residence with a Gas-Fired Dehydration Cooling Unit, by A. P. Kratz, S. Konzo and E. L. Broderick (see p. 203).

The Influence of Physiological Research on Comfort Requirements, by R. W. Keeton, F. K. Hick, N. Glickman and M. M. Montgomery (see p. 159).

A Comparison of the Weight, Particle Count and Discoloration Methods of Testing Air Filters, by F. B. Rowley and R. C. Jordan (see p. 29).

Friction-Heads in Standard Six-Inch Pipe, by F. E. Giesecke and J. S. Hopper (see p. 71).

Following the report of the Committee on Research Mr. Fleisher spoke on the research of the Society and its chairmanship.

The third technical session was held on Tuesday, January 28, at 2:00 p.m. in the ballroom on the Mezzanine of the Muehlebach Hotel, at 2:00 p.m., with E. O. Eastwood, Seattle, Wash., Second Vice-President of the Society, presiding.

Following the presentation of the technical papers and various announcements by Chairman Giesecke, who had resumed the chair, the third technical session adjourned at 5:15 p.m.

The fourth and final technical session was held on Wednesday, January 29, in the Trianon Room, Hotel Muehlebach, and convened at 10:00 a.m. with President Giesecke presiding.

Chairman Giesecke then announced that the total registration at this meeting was 418, including 246 members, 101 guests and 71 ladies. He then called upon N. D. Adams for the report of the Membership Committee. Mr. Adams, in

reporting for the year 1940, expressed his recognition of the splendid cooperation of chapter membership committees and then delivered the report as follows:

Report of Membership Committee

The Membership Committee in reporting for the year 1940, wishes to recognize the splendid cooperation of the Chapter Membership Committees, and the untiring efforts of our Secretary and his assistants.

During the past year we published a pamphlet entitled *The Aims and Activities of the ASHVE,* which was distributed through the Chapters. This was used to inform our present members and help them interest prospective members in the worth-while work of this Society.

We have assisted in the formation of three new Chapters:—Nebraska with headquarters at Omaha; Iowa at Des Moines; and Connecticut at New Haven. The Iowa and Nebraska Chapters were formed by dissolving the old Iowa-Nebraska Chapter, and securing enough new members in each state to form the two separate chapters. This change has made it possible for members in each state to attend more meetings during the year and has increased interest and activities.

The work of the Membership Committee has been handled by dividing the chapters into three groups, western, central and eastern, one member of the committee residing in each section and corresponding with the chapters in his zone.

In obtaining new members during the past year our emphasis has been on quality, rather than quantity. We have been more interested in getting men to join the Society who are first interested in the work of the Society, and who desire to have a part in the service to Society rather than a personal gain. For the calendar year 1940 a total of 274 candidates were elected as follows: Members—97; Associates—105; Juniors—44; and Students—28.

Due to doubtful and economic disturbances during the year we have had an unusually large number of resignations added to those who have been dropped for non-payment of dues. Some of the latter should have been dropped during previous years. The Council accepted 77 resignations, dropped 256 of all grades and 16 were lost by death.

This leaves us with a total membership of 3003 as of December 31, 1940 as compared with 3147 as of December 31, 1939, yet the income from members dues shows an increase over 1939 of $2,936.79. We evidently have dispensed with some dead timber, as this represents a more active and interested membership. The present membership status is:

Honorary Member	1
Presidential Members	20
Life Members	56
Members	1628
Associates	884
Juniors	354
Students	60
Total	3003

Your committee has recommended to the Council that the membership committee, because of its importance to the Society welfare be made a three-year three man committee with one new member appointed each year and the third year member to be made Chairman. This would insure a continuity of work.

Respectfully submitted,

MEMBERSHIP COMMITTEE

N. D. ADAMS, *Chairman*
E. K. CAMPBELL, T. H. URDAHL

Chairman Giesecke announced that the installation of new officers would follow.

New Officers Installed

John F. Hale, Life Member and President of the Society in 1913, conducted the installation. Those inducted into office were, W. L. Fleisher, New York, *President,* E. O. Eastwood, Seattle, Wash., *First Vice-President,* J. H. Walker, Detroit, *Second Vice-President* (who was unable to be present), and M. F. Blankin, Philadelphia, *Treasurer.* The newly elected Council members were, Prof. A. P. Kratz, Urbana, Ill., W. A. Russell, Kansas City, Mo., L. P. Saunders, Lockport, N. Y., and C. Tasker, Toronto, Ont., Canada.

Chairman Eastwood turned the chair over to President-elect Fleisher, who called upon A. V. Hutchinson, secretary, to read the present Constitution and the changes suggested are as follows:

ARTICLE C-II—*Section 4*

A member shall be over thirty (30) years of age and shall have more than eight (8) years' experience in the sciences relating to the arts of heating, ventilating or air conditioning. He shall have been in active practice of his profession and in responsible charge of important work for four (4) years, consisting of design, construction, research, development or teaching, and shall be qualified to design or direct such engineering work.

ARTICLE C-II—*Section 5*

A Junior Member shall be a person over twenty (20) and under thirty (30) years of age who has been actively engaged in the sciences relating to the arts of heating, ventilating or air conditioning work for three (3) years. Each successfully completed year in an engineering school may be considered equivalent to one (1) year of such work.

ARTICLE C-III—*Section 4*

When a Student Member discontinues his regular studies in an engineering college or technical school, he shall be transferred to Junior grade, if he has attained its qualifications, unless he has applied for or has been elected to Associate grade. In no case shall the period of Student Membership exceed four (4) years.

President Fleisher, in response to an inquiry from the floor, explained that by these changes in the Constitution it was hoped to open the Society to a larger group, in other words, instead of saying that a man must have had education in heat transfer before he can become a member of the Society, it is set forth that he must have experience in the sciences relating to the arts of heating, ventilating or air conditioning.

Dr. C.-E. A. Winslow moved the approval of the changes in the By-Laws and the motion was seconded by W. C. Randall, put to a vote and carried.

E. K. Campbell made a recommendation for the Kansas City Chapter that the dues of the Society be reduced, if possible.

President Fleisher said that the resolutions or recommendations would be put on record and be taken up by the Council. He further explained that the Council is already discussing the question of dues and amendments to the By-Laws for continuing the Council authority to 1945 would go to members prior to the Semi-Annual Meeting.

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Report of Resolutions Committee

N. D. Adams, chairman of the Resolutions Committee offered the following resolutions:

WHEREAS, the 47th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been an outstanding success in the City of Kansas City, Mo., January 27-29, 1941,

THEREFORE BE IT RESOLVED THAT an expression of thanks and appreciation be adopted and spread upon the Minutes of the Society and copies thereof be transmitted to each of the persons and agencies which have contributed to make this meeting so enjoyable for the members of the Society, who attended:

To the Kansas City Chapter of the Society and their ladies, who have extended so many courtesies and received us with such gracious hospitality,

To the members of the Committee on Arrangements of Kansas City Chapter for the fine program of entertainment,

To the authors of technical papers for their valuable contributions and discussions,

To the management and staff of the Hotel Muehlebach for their excellent service.

To the Kansas City Visitors and Convention Bureau for its fine cooperation and assistance,

To the Postal Telegraph Co. for its courtesy in furnishing a registration service board,

To the newspapers of Kansas City for their coverage of our sessions and to the trade papers for their excellent publicity and attendance at the meeting,

To the City of Kansas City, TWA, and the various manufacturing concerns, who arranged interesting inspection trips,

To the railroads for their courteous and efficient service,

To Roe Bartle for his inspiring luncheon address and the banquet speaker, Tom Collins, for his entertaining talk,

To the Kansas City Police Quartet for their harmonious musical numbers,

To John Hale and his assistants, who so ably conducted the installation of the new officers, and

To all those who have contributed so much to our pleasure and enjoyment in Kansas City.

Respectfully submitted,
RESOLUTIONS COMMITTEE,
N. D. ADAMS, *Chairman*
M. F. BLANKIN, E. N. MCDONNELL

Chairman Fleisher thanked Mr. Adams, and Professor Eastwood moved that they be approved. The motion was seconded by Mr. Randall, put to a vote and unanimously carried.

At 12:45 noon the final technical session of the 47th Annual Meeting was adjourned.

Entertainment

Members and ladies who arrived early in Kansas City were received by the local committee on Sunday afternoon for a social hour and at 6:30 a buffet supper was served at the Kansas City Club. The response to the invitation to enjoy the hospitality of the Kansas City Chapter resulted in an attendance of 140.

Monday the Kansas City Engineers Club were the hosts at the President Hotel, for a luncheon meeting and the featured speaker was Dr. F. E. Giesecke, president of ASHVE, whose subject was Factors Affecting the Span of Human Life.

During an afternoon social hour in the Music Room of Muehlebach Hotel a program of sound movies in color was given. During the evening the Past

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Presidents had their annual dinner and those in attendance were: W. H. Driscoll, Syracuse, N. Y., John Howatt and John F. Hale, Chicago, W. T. Jones, Boston, Mass., J. I. Lyle, Syracuse, J. F. McIntire, Detroit, and F. B. Rowley, Minneapolis, Minn.

On Tuesday morning the ladies visited the Nelson Gallery of Art and had luncheon at the Junior League Club, after which they went on a motor tour of the country club district, which includes some of the finest residential developments in the city.

At a get-together luncheon in the Trianon room of the Muehlebach Hotel, members and their guests heard an inspiring talk by H. Roe Bartle, Kansas City attorney. During the afternoon some of the members made inspection trips to the Public Auditorium, City Hall, and TWA hangars and other points of interest.

The annual banquet and dance was held in the ballroom of the Muehlebach Hotel at 7:00 p.m. on Tuesday, and a fine menu, which included Kansas City sirloin steak as the featured dish, was greatly enjoyed by the 275 who attended. W. A. Russell, Kansas City Council member, acted as toastmaster and introduced W. H. Driscoll, who presented Dr. Giesecke with the Past President's emblem. W. L. Fleisher, president-elect, was introduced and briefly expressed his appreciation for the honor conferred upon him. A humorous and entertaining address was made by Tom Collins, columnist, Kansas City Journal. After the banquet the toastmaster concluded the program with the introduction of the Officers and Committee on Arrangements. During the dinner vocal numbers were given by the Kansas City Police Quartet, and after dinner a floor show made an enjoyable interlude before the dancing.

PROGRAM 47TH ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
HOTEL MUEHLEBACH—KANSAS CITY, MO.—HOTEL PRESIDENT

TECHNICAL SESSIONS

Monday, January 27

8:30 A.M. Registration (Mezzanine Floor) (*Hotel Muehlebach*)
9:30 A.M. Chapter Delegates Conference (Music Room)
2:00 P.M. TECHNICAL SESSION (Ballroom—Mezzanine) *Hotel Muehlebach*
Reports of Officers and Council Committees
A Comparison of the Weight, Particle Count and Discoloration Methods of Testing Air Filters, by Frank B. Rowley and Richard C. Jordan
Comfort in High Altitude Flying, by D. W. Tomlinson
Boiler Testing Codes, by J. F. McIntire
Warm Air Furnace Testing Codes, by F. L. Meyer
Codes in the Stoker Industry, by E. C. Webb
Report of Tellers of Election, C. S. Pabst, *Chairman*

Tuesday, January 28

10:00 A.M. TECHNICAL SESSION (Ballroom—Mezzanine)
Report of Committee on Research—A. E. Stacey, Jr., *Chairman*
Friction-Heads in Standard Six-Inch Pipe, by F. E. Giesecke and J. S. Hopper

Radiation as a Factor in the Sensation of Warmth, by F. C. Houghten, S. B. Gunst and J. Suciu, Jr.
Operating Results of a Residence Radiant Wall Heating System, by E. J. Rodee

2:00 P.M. TECHNICAL SESSION (Ballroom—Mezzanine)
Comfort Requirements for Low Humidity Air Conditioning, by F. C. Houghten, H. T. Olson and S. B. Gunst
The Influence of Physiological Research on Comfort Requirements, by R. W. Keeton, F. K. Hick, Nathaniel Glickman and M. M. Montgomery
Comfort and Health and Temperature—A Mathematical Solution, by W. L. Fleisher and W. L. Fleisher, Jr.
Summer Cooling in Research Residence with a Gas-Fired Dehydration Cooling Unit, by A. P. Kratz, S. Konzo and E. L. Broderick

Wednesday, January 29

10:00 A.M. TECHNICAL SESSION (Trianon Room)
A Laboratory Method for Cyclic Heat Measurements on Walls and Roofs, by F. G. Hechler and E. R. Queer
Reheating by Means of Refrigerant Compressor Discharge Gas, by S. F. Nicoll
Heat Gain Through Western Windows With and Without Shading, by F. C. Houghten and David Shore
New Business—Installation of Officers—Adjournment.

COMMITTEES

Sunday, January 26

2:30 P.M. Council Meeting (Room No. 1—Mezzanine)
8:30 P.M. Meeting Committee on Research (Room No. 1—Messanine)

Monday, January 27

8:30 A.M. Technical Advisory Committee on Air Distribution and Air Friction (Room 1)
9:00 A.M. Technical Advisory Committee on Insulation (Room 2)
9:00 A.M. Technical Advisory Committee on Solid Fuels (Room 3)
10:00 A.M. Technical Advisory Committee on Radiation and Comfort (Room 4)
10:00 A.M. Technical Advisory Committee on Sensations of Comfort (Parlor A)
10:00 A.M. Committee on Uniform Methods of Heat Loss Calculations (Room 1126)
4:30 P.M. Technical Advisory Committee on Heat Requirements of Buildings (Room 2)
9:00 P.M. Organization Meeting 1941 Committee on Research (Room No. 1—Mezzanine)

Tuesday, January 28

8:30 A.M. Meeting of Nominating Committee (Music Room) Mezzanine
8:30 A.M. Technical Advisory Committee on Instruments (Room 1)
8:30 A.M. Committee on Codes for Testing and Rating Air Cleaning Devices (Room 2)
2:00 P.M. Technical Advisory Committee on Corrosion (Room 1)
4:00 P.M. Technical Advisory Committee on Weather Design Conditions (Room 2)
4:00 P.M. Technical Advisory Committee on Heat Transfer of Finned Tubes with Forced Air Circulation (Room 3)
4:30 P.M. Technical Advisory Committee on Cooling Load in Summer Air Conditioning (Room 4)

Wednesday, January 29

1:30 P.M. Council Meeting (Room No. 1—Mezzanine)

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LADIES' EVENTS

Monday, January 27

9:30 A.M. Get-Acquainted Meeting (Music Room) Mezzanine
12:15 P.M. Luncheon—Kansas City Club (followed by tour)
2:00 P.M. Inspection Tour—Kansas City Museum (Kersey Coates Drive—Cliff Drive—Swope Park)

Tuesday, January 28

10:00 A.M. Inspection Tour—William Rockhill Nelson Gallery of Art (busses leave hotel)
12:15 P.M. Luncheon—Junior League Club
2:00 P.M. Inspection Tour—Country Club District

Wednesday, January 29

10:00 A.M. Shopping Tours—Trips as desired

SOCIAL EVENTS

Sunday, January 26

1:30 P.M. Reception of Members and Ladies (Music Room—Mezzanine)
6:30 P.M. Social Hour and Dinner (Kansas City Club)

Monday, January 27

12:15 P.M. Joint Luncheon with K. C. Engineers Club—*Speaker,* Dr. F. E. Giesecke; *Subject,* Factors Affecting the Span of Human Life—*Hotel President*
4:30 P.M. Sound Movies (Parlor A)
6:00 P.M. Social Hour (Music Room)
6:30 P.M. Past Presidents Reunion and Dinner (Room 3)
7:00 P.M. Informal Get-together Dinner and Entertainment (Terrace Grille)

Tuesday, January 28

12:15 P.M. Get-together Luncheon—Speaker Roe Bartle, Trianon Room
4:30 P.M. Inspection Trips (City Hall—Public Auditorium)
7:00 P.M. Banquet and Dance (Ballroom). Toastmaster, W. A. Russell; Speaker, Tom Collins—Presentation of Past Presidents Emblem

Wednesday, January 29

1:30 P.M. Inspection Trips

COMMITTEE ON ARRANGEMENTS

JOHN M. ARTHUR, *General Chairman*

W. L. CASSELL, *Co-Chairman* — A. D. MARSTON, *Vice-Chairman*

Registration—Henry Nottberg, Jr.
Reception—N. W. Downes
Entertainment—L. A. Stephenson
Housing—F. A. Sheppard
Transportation—E. M. Jolley
Banquet—W. L. Cassell
Finance and Budget—R. P. Campbell
Publicity—F. F. Dodds
Ladies—Mrs. William A. Russell
Seeing-Kansas-City-Committee—Henry E. Gould

No. 1169

A COMPARISON OF THE WEIGHT, PARTICLE COUNT AND DISCOLORATION METHODS OF TESTING AIR FILTERS

By FRANK B. ROWLEY * AND RICHARD C. JORDAN,** MINNEAPOLIS, MINN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Graduate School of the University of Minnesota in cooperation with the Engineering Experiment Station, University of Minnesota.

Part 1

THREE fundamentally different test methods based on weight, particle count, and discoloration have been developed for use in the laboratory rating of air filters. The weight arrestance or efficiency indicates the percentage of the weight of dust in the air which the filter is capable of removing. The particle count arrestance shows the percentage of the number of dust particles which the filter can remove. The discoloration arrestance is a measure of the relative reduction in the potential soiling properties of the air-dust mixture caused by passing it through the filter. Thus, each test method measures a different property, and a filter subjected to all three tests will have three different arrestance or efficiency ratings. Furthermore, any single test method can be used in conjunction with any one of a number of different artificial test dusts possessing different physical and chemical properties, with each dust giving different test results.

In 1933 the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS incorporated the weight principle of testing air filters in their Standard Code for Testing and Rating Air Cleaning Devices used in General Ventilation Work. Several standard dusts have been prepared [1] for use in conjunction with this test method, and several articles [2, 3, 4] dealing with the various factors affecting the test results obtained by the weight method have been published.

* Director, Engineering Experiment Station, University of Minnesota. MEMBER of ASHVE.
** Instructor, Engineering Experiment Station, University of Minnesota. MEMBER of ASHVE.

[1] ASHVE RESEARCH REPORT, No. 1143—A Standard Air Filter Test Dust, by F. B. Rowley and R. C. Jordan. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 681.)

[2] ASHVE RESEARCH REPORT No. 1094—Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity through Filter, by F. B. Rowley and R. C. Jordan. (ASHVE TRANSACTIONS, Vol. 44, 1938.)

[3] ASHVE RESEARCH REPORT No. 1122—Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density, by F. B. Rowley and R. C. Jordan. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

[4] ASHVE RESEARCH REPORT No. 1145—The Effect of Lint on Air Filter Performance, by F. B. Rowley and R. C. Jordan. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

Presented at the 47th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., January, 1941.

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However, the weight test has never received unanimous acceptance by either the filter manufacturers or those individuals interested in filter rating methods, especially since the advent and development of the discoloration test method. For this reason there has been a need for extended studies covering a direct comparison and correlation of these three test methods. The present investigations cover the following points:

1. Factors affecting the results obtained by the discoloration test method.
2. Comparison of the weight, particle count, and discoloration methods of test on typical filters with various test dusts.
3. Correlation of filtering theory with test results.

WEIGHT TEST METHOD

The weight or crucible method of determining filter arrestance as used in these investigations was essentially that described in the ASHVE Air Filter Code.[5] The test apparatus and procedure have been described in a previous paper.[6] The rate of dust feet was 40 g (grams) per hour and the face air velocity 300 fpm. In the majority of tests determinations were made only of the arrestance, as filter life is the same by all three test methods.

PARTICLE COUNT METHOD

The particle count method of determining filter arrestance consists in removing air samples simultaneously from upstream and downstream of the filter and determining by microscopic examination the number of particles per unit volume of air. Equation (1) can then be used to determine the arrestance:

$$\text{Arrestance} = 1 - \frac{P_1}{P_2} \qquad (1)$$

where, P_1 = dust count downstream of filter, particles per cubic foot.
P_2 = dust count upstream of filter, particles per cubic foot.

In the present investigation these air-dust samples were collected by two different methods; *first,* by means of an adhesive impingement type of dust counter developed in the University of Minnesota, Engineering Experiment Station, for use in making atmospheric dust count determinations; and *second,* by means of dust settling chambers. The dust counter used, shown in Fig. 1, consists of a suction pump which draws the air-dust mixture through a small orifice and impinges it against a glass slide coated with a thin film of oil. During sampling a vacuum is maintained in the impingement chamber at approximately 5 in. of mercury by means of the pump. The impingement slide carriage is linked to the pump handle so that during operation the slide moves slowly past the orifice. Thus, the jet of air and dust is impinged at constant velocity against the oiled slide, and the dust particles are deposited along a path. By microscopic examination of this path at 100 diam magnification the number of particles per unit volume of air can be determined.

The settling chambers used in these tests were similar to those described by Green.[7] In the present investigation the chambers were 4 in. x 4 in. in cross-

[5] ASHVE Standard Code for Testing and Rating Air Cleaning Devices used in General Ventilation Work. (ASHVE TRANSACTIONS, Vol. 39, 1933.)
[6] Loc. Cit. Note 2.
[7] Some Accurate Methods of Determining the Number and Size Frequency of Particles of Dusts, by H. L. Green. (*Journal of Industrial Hygiene,* Vol. 16, January, 1934.)

section and 2 in. in height. Samples of the air-dust mixture to be analyzed were trapped in these chambers and the dust allowed to settle out of the air for 24 hours onto oil-coated glass slides placed on the chamber bottoms. By microscopic examination and count at 100 diam magnification the number of particles per unit volume of space was determined. The settling time was sufficient to insure complete removal from the air of all suspended particles visible at this magnification.

It was found impractical to use a high rate of artificial dust feed to the filters when making particle count arrestance determinations because of the resulting extremely high dust counts. Therefore, these tests were conducted with either normal air-borne dust and no artificial dust feed, or by dusting the

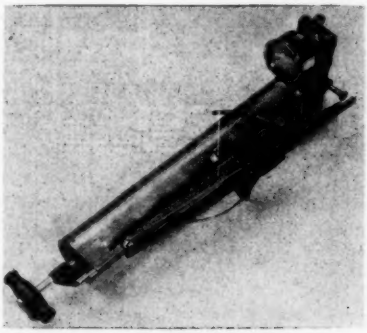

FIG. 1. DUST COUNTER USED IN MAKING PARTICLE COUNT ARRESTANCE DETERMINATIONS

air in the vicinity of the test duct inlet with the desired artificial dust. This dusting was accomplished about 10 ft away from the duct entrance by means of a compressed air jet and a small container of the test dust. In this way it was possible to obtain concentrations sufficiently low that they could be readily counted yet high enough that their mixture with the normal air-borne dust did not introduce any appreciable error. As the upstream and downstream samples were taken simultaneously, small variations in the dust concentrations had no appreciable effect on the results.

In making both the settling chamber and the dust counter arrestance determinations five samples taken from each side of the filter were counted and averaged for a single arrestance determination. The arrestance results reported are the average of five separate determinations or tests.

DISCOLORATION TEST METHOD

The discoloration method (also sometimes termed photo-electric cell, optical density, or soiling-power method) of determining filter arrestance consists essentially in removing air samples simultaneously from upstream and down-

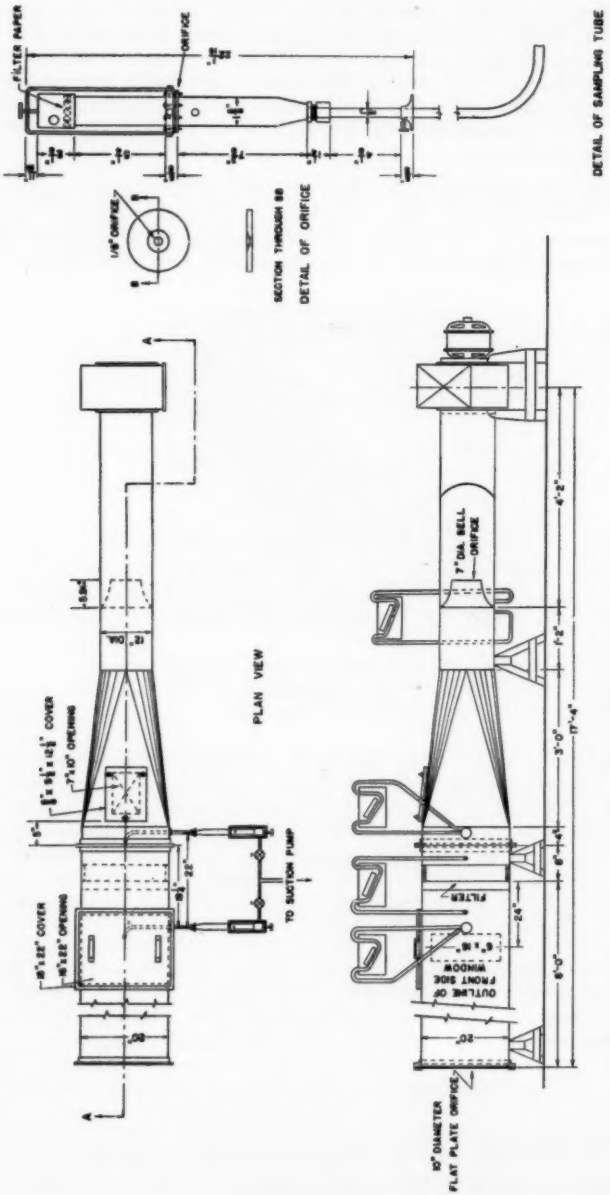

FIG. 2. TEST DUCT WITH SAMPLING APPARATUS FOR DETERMINING DISCOLORATION ARRESTANCES

stream of a filter, passing these samples through filter papers to remove the dust, and then subjecting these papers to a light beam and photo-cell to determine the concentrations of the dust deposits. The arrestance is determined by application of Equation (2).

$$\text{Arrestance} = 1 - \frac{C_1}{C_2} \quad \ldots\ldots (2)$$

where, C_1 = downstream dust concentration.
C_2 = upstream dust concentration.

The density of dust on the filter paper D will vary with the area, the rate of sampling, the concentration, and the time of sampling as in Equation (3):

$$D = \frac{QtC}{A} \quad \ldots\ldots (3)$$

where, Q = volume of air sampled, cubic foot per minute.
t = time of sampling, minutes.
A = area of dust deposit, square inch.
C = concentration of dust in air stream.
D = density of dust deposit on filter paper.

If Equation (3) is solved for C then,

$$C = \frac{DA}{Qt} \quad \ldots\ldots (4)$$

This expression may then be substituted in the arrestance equation, giving

$$\text{Arrestance} = 1 - \frac{D_1 A_1 Q_2 t_2}{D_2 A_2 Q_1 t_1} \quad \ldots\ldots (5)$$

In order to apply this relationship in practice three of the four variables D, A, Q and t should be kept constant and the other varied. As it is difficult to measure D in absolute units but is comparatively easy to determine when D_1 and D_2 are equal, this variable is usually adjusted so that D_1 and D_2 cancel.

The apparatus used in the present investigations was essentially the same as that developed and described by Dill [8] and used at the National Bureau of Standards. However Dill in his test made

$$D_1 = D_2$$
$$t_1 = t_2$$
$$Q_1 = Q_2$$

and thus,

$$\text{Arrestance} = 1 - \frac{A_1}{A_2} \quad \ldots\ldots (6)$$

In this case the size of the filter paper spot is varied by providing various apertures corresponding to the different arrestances. In order to provide more flexibility in the tests of the present investigations the apparatus was modified so that,

$$D_1 = D_2$$
$$t_1 = t_2$$
$$A_1 = A_2$$

and thus,

$$\text{Arrestance} = 1 - \frac{Q_2}{Q_1} \quad \ldots\ldots (7)$$

Thus in these tests the volumes of air sampled from the two sides of the filter were varied until the resulting dust spot densities of the filter papers were equal.

[8] A Test Method for Air Filters, by R. S. Dill. (ASHVE TRANSACTIONS, Vol. 44, 1938.)

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Fig. 2 shows the test duct with the sampling tubes in position and a detail of these tubes. The air samples were drawn from the test duct by means of a suction pump and the volumes of these samples measured by means of calibrated flat plate orifices located in the sampling tubes. The pressure differentials across these orifices were measured by means of inclined draft gages. The rate of sampling for the downstream tube was 0.75 cfm and the upstream rate was varied, depending on the arrestance. The time of sampling varied from two to ten minutes, depending upon the nature and color of the test dust. The photometer circuit used for determining when the densities of the filter paper dust deposits were the same was identical with that used by Dill and originally described by Wood.[9]

There are several methods by which the photometer can be used to determine when the dust-spot densities of the filter papers are equal. The method originally described by Dill placed the filter paper dust spot between the light

FIG. 3. PHOTOMETER FOR TRANSMITTED LIGHT DISCOLORATION ARRESTANCE DETERMINATIONS

source and the photronic cell, and thus measured the intensity of light transmitted by the filter papers. In addition to arrestances determined by this method, the present investigations also include arrestances determined by measuring the light reflected from the surfaces of the filter papers. Fig. 3 shows the apparatus used for making transmitted light discoloration arrestance determinations, and Fig. 4, the apparatus for making reflected light discoloration arrestance determinations. In the latter case both the incident and reflected light beams were at angles of 45 deg to the filter paper surface, and at 90 deg to each other.

FACTORS AFFECTING DISCOLORATION TEST RESULTS

During the course of the investigations it was found that several factors influenced the arrestances determined by the discoloration test method. Each of these factors was subjected to further analysis and will be discussed in turn.

[9] Zero-Potential Circuit for Blocking-Layer Photo-cells, by L. A. Wood. (*The Review of Scientific Instruments*, Vol. 7, March 1936.)

Velocity of Air through Sampling Tubes

At the beginning of the investigations it was realized that when the sampling tube air volumes are varied in order to obtain equal discolorations of the upstream and downstream filter papers, there are necessarily concomitant variations in the sampling tube air velocities. This results in two possible conditions, one which the sampling tube air velocity is greater than the duct air stream velocity, and the other in which the tube velocity is lower than the duct velocity. With the latter condition there is no apparent reason why the air-dust mixture samples should not be representative of those at the point of sampling, but when the tube velocity is greater than the duct velocity, not only the jet of air directly in front of the tube is being sampled, but also air surrounding this core. It would then appear that, since the flow of air-dust mixture must change directions in order to enter the sampling tube, some of the dust may be centrifuged out of the air stream and not taken into consideration.

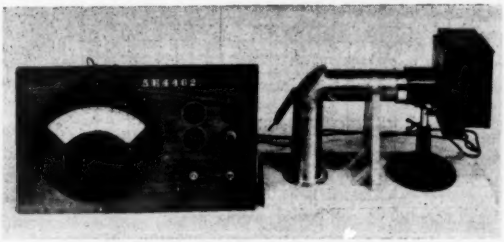

FIG. 4. PHOTOMETER FOR REFLECTED LIGHT DISCOLORATION ARRESTANCE DETERMINATIONS

In order to determine the possible magnitude of this source of error a series of filter tests was run in which the sampling velocities were varied. In some cases the opening sizes of the sampling tubes were changed by the application of two tips, one large and one small, placed over the ends of the main sampling tube. The large tip was 0.667 in. in diameter, and the small one 0.182 in. in diameter, and both were 3 in. in length. These permitted extreme entrance velocity variations without any change in air volume. Normally the highest velocity used is approximately 718 fpm, with a duct air velocity of 300 fpm. In these tests the velocity was varied between 2,990 and 256 fpm.

The results of these tests indicated that any errors in the test results introduced by normal variations in the sampling velocities were negligible. For example, one filter tested three different times with downstream tube air velocities of 718 fpm, 574 fpm, and 383 fpm showed identical results. Only when the sampling tube velocities were increased to three or four times normal was there any significant change in the resulting filter arrestance. For example, on one sample when the upstream sampling tube air velocity was increased from 536 fpm to 2,990 fpm the transmitted light discoloration arrestance also increased from 25 to 28 per cent.

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Selective Filtering

As some types of dust are more easily removed from the air than others, and large dust particles are more easily eliminated than small ones, most air filters exhibit the phenomenon of selective filtering. As a result dust samples drawn from either side of a filter and deposited on filter papers may exhibit the same photronic cell response to either reflected or transmitted light and yet appear different in color to the human eye. This was noticeable in discoloration tests in which the artificial test dust was either a mixture of two different dusts (such as a mixture of 80 per cent Pocahontas ash and 20 per cent carbon dust) or a single component dust of screened ash (such as Pocahontas ash). In the case of the mixed dust this was probably caused by a higher filtering efficiency for one of the components than for the other, thus

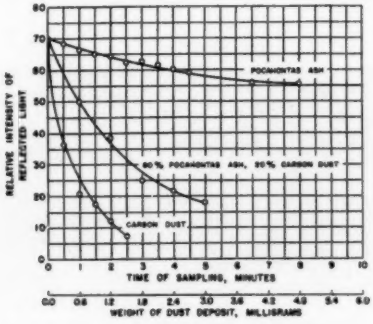
RELATIVE INTENSITY OF REFLECTED LIGHT
POCAHONTAS ASH
80% POCAHONTAS ASH, 20% CARBON DUST
CARBON DUST
TIME OF SAMPLING, MINUTES
WEIGHT OF DUST DEPOSIT, MILLIGRAMS

FIG. 5. EFFECT OF DUST DEPOSIT WEIGHT ON INTENSITY OF REFLECTED LIGHT

resulting in a different mixture leaving the filter than that fed to the filter. In the case of the tests made with straight ash dust it appeared that there was always some carbonaceous material in the ash which again would result in a selective filtering and a visible difference in the appearance of the matched papers. The only cases in which the photronic cell matched papers also appeared perfectly matched to the human eye were those in which single component dusts relatively free from any impurities of a different color were used. Those dusts tested which most closely fill these requirements were carbon dust and Cottrell ash.

Papers Matched by Transmitted and Reflected Light

As previously noted discoloration arrestance determinations were made by two different methods, one in which the dust deposits on the filter paper were considered to be of equal density when the photronic cell response to light transmitted through these papers was the same, and the other when the photronic cell response to light reflected from the surfaces of the papers was the same. These two methods of test gave entirely different arrestance determinations, as dust deposits matched by one method were not matched by

the other method. The results of these comparative tests will be presented later in the paper.

It appears that if air filter arrestance ratings are to be based upon the relative discoloration powers of air-dust mixtures before entering and after leaving the air filter that the measure of this discoloration should be in direct relation to the response of the human eye. As all surface discolorations observed by the human eye are received by reflected light it seems justifiable that this principle should also be used in measuring the discoloration of the filter papers.

As discoloration is usually defined in terms of the human eye's response, theoretically the most accurate results will be obtained when the spectral sensitivity of the photronic cell is equivalent to that of the human eye. The ordinary photo-electric cell is considerably more sensitive than the eye in the long and short wave length regions of the visible spectrum, and furthermore its sensitivity extends into the intra-red and ultra-violet regions. In order to determine what effect this had on the reflected light discoloration arrestance determinations several tests were run in which a correction filter was applied to the photronic cell in order to duplicate the spectral sensitivity of the human eye. There were no noticeable differences in the results obtained with and without this correction filter with the test dusts and light source used in these investigations, but it is conceivable that this might not always be the case.

Sensitivity of Discoloration Test Apparatus

The sensitivity of the photometer in detecting variations in the volumes of air sampled is dependent upon the amount of dust deposited on the filter papers. If the dust concentrations on the papers are heavy, additional dust deposits may result in little change in the photo-cell response. For this reason tests were made with both transmitted and reflected light to determine the change in photometer readings per unit increase in dust deposit, and from these tests it was possible to deduce the optimum time of sampling for maximum sensitivity. Fig. 5 shows typical response curves for carbon dust, Pocahontas ash, and a mixture of 80 per cent Pocahontas ash and 20 per cent carbon dust. These tests were made at a sampling rate of 0.75 cfm and an air-dust concentration of 1 g of dust in 1250 cu ft of air. As would be expected, the darker the color of the dust the more rapidly the meter reading dropped off. These curves indicate that when carbon dust is used as a test dust, the time of sampling should be 1 to 2 min, whereas, with Pocahontas ash the sampling time might be increased to as much as 4 or 5 min. If in the case of the carbon dust the sampling time was increased to 3 min or more the photometer response would be approaching a minimum, and it would be difficult to detect slight changes in the sampling rate. The response curves for transmitted light are similar to those shown in Fig. 5 for reflected light with the exception that the photometer readings approach zero when the dust deposits are heavy enough to completely block out all transmitted light instead, as in the case of the reflected light, of approaching a constant photometer reading greater than zero when the papers are completely coated with the test dust.

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Relative Soiling Power of Different Test Dusts

Both the transmitted and reflected light methods of determining filter arrestance actually measure the reduction in soiling power or discoloration only in relation to the test dust being used. Thus, it is possible for a filter to show the same arrestance when tested on either a dark dust or a light colored dust, although the absolute soiling power of the dark dust per unit volume or weight would be many times greater than that of the light colored dust. Thus, it is conceivable that a filter tested by either the transmitted or reflected light methods might pass 80 per cent of a light colored dust on one test and 20 per cent of a dark colored dust on another, and yet this 80 per cent might not have nearly the potential soiling power that the 20 per cent of the dark colored dust would have. The variations in the absolute soiling powers of three different test dusts are indicated by Fig. 5, in which the reflected light photometer response is shown for identical dust deposits of Pocahontas ash, carbon dust, and the 80-20 mixture. Thus, for example, with a dust deposit of 1.6 milligrams the 80-20 mixture reflects 4.5 times and the Pocahontas ash 9 times as much light as the carbon dust. The necessity for a standard test dust to be used in conjunction with discoloration tests is apparent from these considerations.

Part 2

Comparison of Test Results

Weight, Particle Count, and Discoloration Methods

It is apparent that filter test results obtained by the three methods (weight, particle count, and discoloration) will not be the same, as each method measures a somewhat different characteristic of the filter. In order to obtain a direct comparison of filter performance with different types of dust and with the three methods of test, four filters were selected as typical of those used in practice and were subjected to comparative tests. These filters were designated by the letters, *A*, *B*, *C*, and *D* and briefly may be described as follows:

A—A permanent type of cleanable oil filter 4 in. thick with 24 layers of expanded metal and wire screen graded from coarse mesh at entrance to fine mesh at leaving side.

B—A viscous coated throw-away type filter 2 in. thick. The fibrous media were graded in fiber size, density, and oiling from entering to leaving side.

C—A cellular type filter 2 in. thick built in two sections, each with the axis of cells set at 45 deg to the center-line of duct and at 90 deg to each other. The cells on the entering side were of larger dimensions than those on the leaving side of the filter.

D—A filter of cotton media of coarse material on the entering side and glazed on the leaving side. The filter media were accordion plaited in frame to give an area of approximately 12 times the cross-sectional area of air stream.

These filters were selected because they were typical of their classifications.

TABLE 1—COMPARISON OF ARRESTANCES BY WEIGHT, DISCOLORATION, AND PARTICLE COUNT METHODS

TEST DUST	ARRESTANCE, FILTER *A*					ARRESTANCE, FILTER *B*					ARRESTANCE, FILTER *C*					ARRESTANCE, FILTER *D*				
	Wt.	Particle Count		Discoloration		Wt.	Particle Count		Discoloration		Wt.	Particle Count		Discoloration		Wt.	Particle Count		Discoloration	
		Dust Counter	Settling Chamber	Trans. Light	Reflected Light		Dust Counter	Settling Chamber	Trans. Light	Reflected Light		Dust Counter	Settling Chamber	Trans. Light	Reflected Light		Dust Counter	Settling Chamber	Trans. Light	Reflected Light
Pocahontas Ash.....	93	..	..	25	51	96	..	33	36	48	85	..	22	18	30	90	..	40	48	67
Cottrell Ash.....	93	..	..	25	..	95	..	..	44	..	83	..	..	19	..	88	..	..	48	..
80 per cent Pocah. 20 per cent Carbon..	85	..	..	7	37	88	..	..	32	47	72	..	..	4	24	99	..	..	53	63
Carbon Dust....	45	..	..	10	37	63	..	..	33	54	33	..	..	10	9	84	..	..	71	71
Normal Air-Borne Dust....	..	..	..	3	11	..	16	16	3	12	..	7	12	3	9	..	23	27	10	16

It should not be construed that they are necessarily superior to other filters of similar design.

Four synthetic dusts were selected for use in these comparative tests, namely: Pocahontas ash, screened through 200 mesh; Cottrell ash, screened through 200 mesh; Double Bolted Carbon Dust[10] screened through 100 mesh; and a mixture consisting of 80 per cent Pocahontas ash and 20 per cent carbon dust by weight. In addition to tests on these synthetic dusts some discoloration and particle count arrestances were obtained using normal air-borne dust without any artificial dust feed.

Filters *A*, *B*, *C*, and *D* were all tested by the transmitted light discoloration and the weight methods on Pocahontas ash, Cottrell ash, carbon dust, and the 80-20 dust mixture. Reflected light discoloration arrestances were obtained on all filters with Pocahontas ash, carbon dust and the 80-20 mixture. In addition

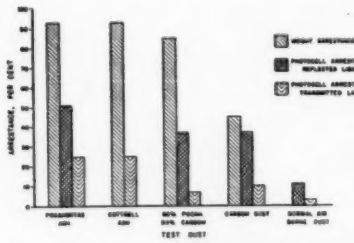

FIG 6. COMPARISON OF WEIGHT AND DISCOLORATION ARRESTANCES FOR FILTER *A* WITH VARIOUS TEST DUSTS

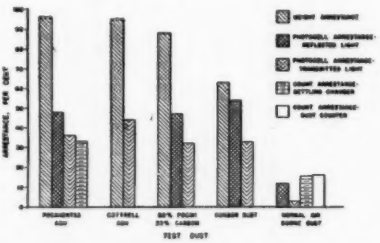

FIG. 7. COMPARISON OF WEIGHT, PARTICLE COUNT AND DISCOLORATION ARRESTANCES FOR FILTER *B* WITH VARIOUS TEST DUSTS

all four filters were tested by both the transmitted and reflected light discoloration methods using normal air-borne dust. Particle count arrestances were determined for Filters *B*, *C*, and *D* by both the dust counter and settling chambers on normal room air dust suspensions and with the settling chambers using Pocahontas ash dusted near the duct inlet.

The results of these tests are shown in Table 1 and in the bar diagrams of Figs. 6, 7, 8, and 9, and indicate a wide variation in arrestances dependent upon both the type of test dust and the test method. In all cases the weight arrestances are higher than either the transmitted or reflected light discoloration arrestances or the particle count arrestances. In practically all cases the relative arrestances obtained by the different methods of testing were in the following order ranging from highest to lowest: weight arrestance, reflected light discoloration arrestance, transmitted light discoloration arrestance, and particle count arrestance.

In this ranking the dust counter and settling chamber arrestances have been grouped together under particle count arrestance as there are no fundamental differences between these two determinations. The only reversals in order were between the discoloration and the particle count test methods for Filters

[10] Loc. Cit. Note 1.

B and *D* on normal air and for Filter *C* on Pocahontas ash, and between the reflected and transmitted light discoloration methods for Filter *C* on carbon dust. Two of these four reversals were on normal air-borne dust and since, in these cases, all of the arrestances were low and grouped together with little spread, and the rate of dust feed was necessarily too low for good accuracy on the discoloration tests, these reversals were probably not reliable and were not used in determining the rank in relative arrestance for the different methods.

In the particle count arrestance tests made on the various filters with normal air-borne dusts, results obtained with the settling chambers check reasonably well with those obtained by means of the dust counter. Both of these methods required a large number of tests in order to obtain a statistically accurate average arrestance. The discrepancies noted between the two methods of test were well within the range of experimental error. The particle count arrest-

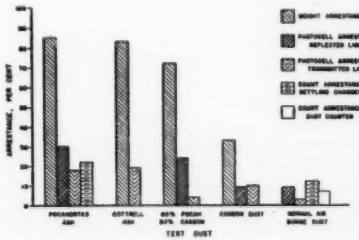

FIG. 8. COMPARISON OF WEIGHT, PARTICLE COUNT AND DISCOLORATION ARRESTANCES FOR FILTER *C* WITH VARIOUS TEST DUSTS

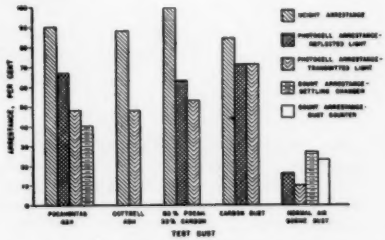

FIG. 9. COMPARISON OF WEIGHT, PARTICLE COUNT AND DISCOLORATION ARRESTANCES FOR FILTER *D* WITH VARIOUS TEST DUSTS

ance tests made on Filters *B*, *C*, and *D* with the settling chambers showed much higher arrestances when the test air was dusted with the 200-mesh Pocahontas dust than when normal laboratory air was used. This result would be expected in view of the larger particle size of the artificial dust.

It is interesting to note that for Filters *A* and *C* on transmitted light, and Filters *B* and *D* on reflected light, the discoloration arrestances are higher for both the carbon dust and the Pocahontas ash tested separately than for the 80-20 mixture of the two components. It is difficult to offer a satisfactory explanation for this without further investigation, but it may be linked to the selective filtering phenomenon previously discussed or to a change in the physical characteristics of the aggregate over those of either of its constituents.

Both the discoloration and the particle count arrestances made using normal air dust suspensions showed extremely low results on all filters. This was probably caused by the higher ratio of the number of small particles to the number of large particles for the normal air-borne dust in comparison with the same ratio for the artificial dust. Filter *D* showed the highest arrestances by all test methods when using these normal dust suspensions.

The following conclusions may be drawn from these investigations:

1. For the majority of tests the relative magnitude of the filter arrestances determined by the weight, particle count, and discoloration methods were in the following

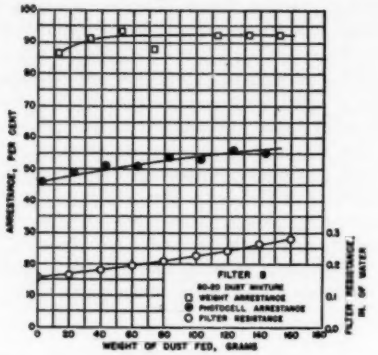

FIG. 10. COMPARISON OF WEIGHT AND DISCOLORATION PERFORMANCE CURVES FOR FILTER *B*

order ranging from highest to lowest: weight arrestance, reflected light discoloration arrestance, transmitted light discoloration arrestance, and particle count arrestance.

2. The particle count arrestances determined by the dust counter method checked well with those determined by the settling chamber method. However, both methods required that a large number of tests be made in order to obtain a statistically accurate average determination.

3. Tests made by using normal air-borne dust resulted in the lowest arrestance values with each of the test methods. This was probably caused by the greater percentage of very small particles found in normal air-dust suspensions than in any of the artificial dusts.

4. In several instances the discoloration arrestances for Pocahontas ash and for carbon dust were both higher than for the 80-20 mixture using these same dusts as components. The exact reason for this is unknown, but it may be caused by a

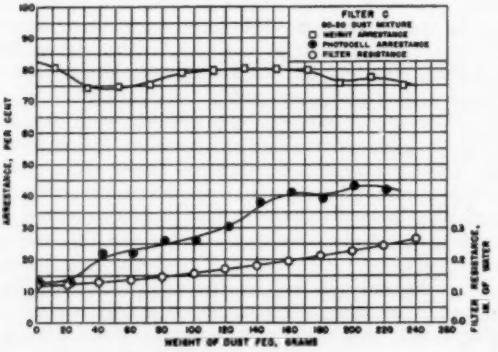

FIG 11. COMPARISON OF WEIGHT AND DISCOLORATION PERFORMANCE CURVES FOR FILTER *C*

change in the physical characteristics of the aggregate over those of either of its constituents.

CONTINUOUS DISCOLORATION FILTER TESTS

In order to determine the effect of dust loading upon the arrestance of air filters tested by the reflected light discoloration method, and also to gain a direct comparison of the weight and reflected light arrestances, continuous tests were conducted on Filters *B*, *C*, and *D* with determinations made by the weight and discoloration methods. The 80-20 dust mixture was used as a test dust, the rate of feed was 20 g per hour, and the filter face air velocity 300 fpm. At the beginning of each test hour the discoloration arrestance was determined

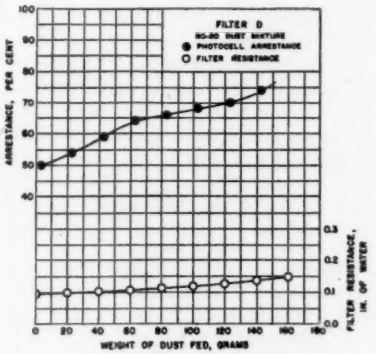

FIG. 12. DISCOLORATION PERFORMANCE CURVES FOR FILTER *D*

after which the dust plate was removed and re-weighed and the weight arrestance determined on the remaining dust. The discoloration arrestance determination consumed approximately 5 out of the 20 g of dust, and the weight arrestance was determined on the remaining portion Filter *D* was tested only by the reflected light discoloration method as the weight arrestance was so high that determinations of the increase in weight of the dust crucible were difficult when based upon a dust feed of only 15 g weight.

The results of these tests shown in Figs. 10, 11 and 12 indicate in all cases a marked rise in reflected light discoloration arrestance with increased weight of dust feed. The arrestance of Filter *D* rose from 50 to 75 per cent during an increase of filter resistance of only 0.05 in. of water. Filter *C* rose in discoloration arrestance from 13 to 43 per cent during an increase in filter resistance of 0.15 in. of water, while during the same time the weight arrestance remained substantially constant. Filter *B* rose in discoloration arrestance from 46 to 55 per cent during a resistance rise of 0.12 in. of water, while the weight arrestance rose only 5 per cent. These tests indicate that, when continuous tests are made on air filters with the arrestances determined by the

discoloration method, much greater variations can be expected than when the arrestances are determined by the weight method. Although none of these tests were conducted over any definite increase in pressure drop across the filter, they indicate that continuous tests can be used to determine the life and arrestance of air filters by the discoloration method. However, the determination of the dust holding capacity by such a test would not be possible.

CORRELATION OF FILTERING THEORY WITH TEST RESULTS

It is apparent from these test results that the three methods of rating filters (weight, discoloration, and particle count) are not comparable and each method measures a somewhat different characteristic of the filter. The particle count method of rating air filters does not distinguish between large and small particles but gives equal emphasis to both. The emphasis given to different sizes of particles by the weight method is in direct relation to the cube of the diameter of the particle as long as the density remains constant. For the discoloration method there is probably no simple relationship between relative effect and size of particle, but it would appear that the projected surface area would be one of the dominating factors. Thus, in this case, the relative effect would be at least partially in relation to the square of the diameter of the particle. If these relationships are true, when tested by the weight method a 10 μ (micron) particle has 1,000 times as much effect on the ultimate rating as a 1 μ particle, and when tested by discoloration the relationship is 100 to 1. When rated by the particle count method both the 10 and 1 μ particles have equal effect.

If air filters were capable of eliminating both large and small dust particles with equal ease there would be little difference between the ratings obtained by the different methods. However, that such is not the case is indicated by Fig. 13 which shows the relationship between size of dust particle and filtering arrestance for three different filters. These results were obtained by the weight method of testing, using a test dust consisting of Pocahontas ash, separated into different size fractions by means of a Federal Pneumatic Air Classifying Unit, the actual particle size being determined by microscopic examination. In these curves arrestance has been plotted against *diameter in microns of dust particle with average volume* since this statistical diameter is also the measure of the dust particle with average weight if the density remains constant. These curves indicate a definite dropping off in arrestance for the smaller sizes of particles for both Filters *B* and *C*. Filter *D*, which incorporated cotton fiber filtering media, showed no noticeable reduction in arrestance within the dust size fractions used in these tests, but very possibly there would have been a lowering of arrestance if it had been possible to extend the test down to ash particles of finer size.

Air-borne dusts are apparently constituted principally of particles of small size. We may therefore assume a hypothetical case in which a filter is 90 per cent efficient on particles of 10 μ diameter and 10 per cent efficient on particles of 1 μ diameter, and the test dust to which the filter is subjected consists of 99 per cent by number of 1 μ particles and 1 per cent of 10 μ particles. If the filtering arrestances are then calculated by the weight, particle count, and discoloration methods of test applying the assumptions previously discussed it is found that the count arrestance is 11 per cent,

the discoloration arrestance 50 per cent, and the weight arrestance 83 per cent. This coincides in general magnitude with the actual test results previously reported with the exception that the discoloration arrestance calculated here is higher. Actually the discoloration arrestance will be dependent principally upon the projected area only when transmitted light and opaque dust particles are used. Under actual operating conditions such factors as the opacity of the dust particles in the case of transmitted light and the relative reflectivity of dust particles in relation to that of the filter paper in the case of reflected light will be equally as important as the projected areas.

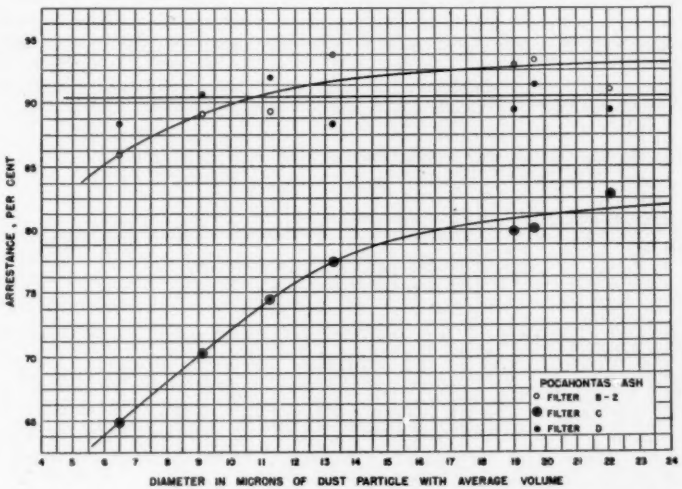

FIG. 13. EFFECT OF DUST PARTICLE SIZE ON WEIGHT ARRESTANCE FOR FILTERS *B*, *C*, AND *D*

In order to determine how high a transmitted light discoloration arrestance could be obtained when using filters based upon mechanical principles of operation, a special filter was constructed and tested. This filter consisted of a Filter *B* of stock construction operated in tandem with 14 layers of oiled cheese cloth supported on a wood frame. When tested with 200-mesh Pocahontas ash the arrestance of this combination by the transmitted light discoloration test was 68 per cent. This same filter combination when tested by the weight method on the same test dust had an arrestance of approximately 94 per cent. When the 14 layers of cheese cloth were removed from the back of the filter and the weight and discoloration arrestances re-run, the weight arrestance was still 92 per cent, whereas the discoloration arrestance dropped to 36 per cent. Thus the addition of the cheese cloth screen in tandem operation with this filter raised the transmitted light discoloration arrestance 32 percentage units, while the weight arrestance remained substantially constant. This test clearly indicates the independence of the weight method and the

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dependence of the discoloration method upon the elimination of the small particles contained in the dust suspension.

The results of all tests indicated that filters based upon mechanical separation of the dust from the air removed the large dust particles much more easily than the small ones. In order to verify further this selective filtering action samples of normal air and of air discharged from filters were obtained in settling chambers and the dust allowed to settle onto glass slides for a period of 24 hours. These slides were then subjected to microscopic examination and the diameters of from 300 to 500 particles per sample recorded. From these data, assuming spherical particles, the average diameter, the diameter of the particle with the average surface area, and the diameter of the particle with the average volume were determined. These statistical diameters were calculated by use of the following Equations (8), (9), and (10):

$$\text{Average diameter} = \frac{\Sigma\,(FD)}{\Sigma\,F} \quad \ldots\ldots (8)$$

$$\text{Diameter of particle with average surface area} = \left[\frac{\Sigma\,(FD)^2}{\Sigma\,F}\right]^{1/2} \quad \ldots\ldots (9)$$

$$\text{Diameter of particle with average volume} = \left[\frac{\Sigma\,(FD)^3}{\Sigma\,F}\right]^{1/3} \quad \ldots\ldots (10)$$

where,

F = dust particle group frequency.
D = average diameter of group limit.

The results of these tests are recorded in Table 2 for air discharged from Filters *B*, *C*, and *D*, and for two samples of unfiltered room air.

These results indicate a definite preferential filtering of the air for all three filters. The diameter of the particle with average volume was reduced from approximately 17 μ to 9.6 μ for Filter *D*, 10.9 μ for Filter *B*, and 11.0 μ for Filter *C*. There were corresponding reductions in the average diameters and in the average surface area diameters.

The particle count arrestances (Table 1) and the particle size distribution data (Table 2) may be used to calculate weight arrestances as a check on the

TABLE 2—COMPARISON OF SIZE OF DUST PARTICLES ON ENTERING AND DISCHARGE SIDES OF TYPICAL AIR FILTERS

SOURCE OF DUST SAMPLE	AVERAGE DIAMETER, MICRONS	DIAMETER OF PARTICLE WITH AVERAGE SURFACE AREA, MICRONS	DIAMETER OF PARTICLE WITH AVERAGE VOLUME, MICRONS	PER CENT PARTICLES BY NUMBER SMALLER THAN:		
				6μ Diameter	18μ Diameter	30μ Diameter
Laboratory Air...	7.2	11.3	16.1	62.9	91.5	97.0
Laboratory Air...	9.6	13.8	18.4	48.1	83.5	95.4
Air Downstream from Filter *B*...	3.5	6.8	10.9	89.0	96.9	98.5
Air Downstream from Filter *C*...	4.7	7.6	11.0	79.3	95.0	99.0
Air Downstream from Filter *D*...	3.8	6.6	9.6	86.8	96.5	98.7

TABLE 3—COMPARISON OF CALCULATED AND TEST WEIGHT ARRESTANCES

FILTER	VOL. AVG. PARTICLE ENTERING FILTER, CU MICRONS	VOL. AVG. PARTICLE LEAVING FILTER, CU MICRONS	ACTUAL TEST ARRESTANCE, PER CENT		CALCULATED WEIGHT ARRESTANCE, PER CENT
			Count	Weight	
B	2716	676	16	88	79
C	2716	684	7	72	76
D	2716	505	22	94	86

actual weight arrestance test values. For example, the volume of particle with average volume upstream of Filter *B* is then 2716 cu microns and downstream is 676 cu microns. As the dust counter particle arrestance was 16 per cent the calculated weight arrestance is then,

$$\text{Arrestance} = 100\left[\frac{2716 - 676\,(1 - 0.16)}{2716}\right] = 79 \text{ per cent} \quad \ldots\ldots\ldots (11)$$

Table 3 shows a comparison of the calculated weight arrestances with actual test weight arrestance values. The actual test values recorded for weight arrestance were for the first hour of test using an 80-20 dust mixture. This initial arrestance was chosen because the dust size distribution data were also determined on unloaded filters. It should be noted that the calculated weight arrestances are all based upon data determined using normal suspensions of dust in room air and that the actual weight arrestance values were determined with an artificial dust mixture. However, previous tests [11] have shown that weight arrestances made on normal air-borne dust check quite closely with those for this artificial mixture.

The reasonably close check for all three filters of the calculated weight arrestance and the actual test weight arrestance values along with the data showing the reduction in size of the average particle upon passage through the filters verify the evidence previously presented as to the selective filtering of mechanical filters.

The following conclusions may be drawn from these investigations:

1. Tests conducted by the weight method using Pocahontas ash of different particle size fractions indicate a definite preferential filtering of the larger particles over the smaller ones.

2. Theoretically the emphasis given to different sizes of particles by the weight method varies as the cube of the diameter, whereas the particle count method gives equal emphasis to all sizes. The effect of particle size upon discoloration arrestance probably varies as the square of the diameter, although undoubtedly other factors are of equal importance in this case.

3. Tests made using 200-mesh Pocahontas ash on a stock filter with and without a series of fourteen oiled cheese cloth screens in tandem showed no appreciable differences in the weight arrestances, but a difference of 32 percentage units in the transmitted light arrestances. This indicates the independence of the weight test

[11] Loc. Cit. Note 3.

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results and the dependence of the discoloration test results upon the small particles when the test dust is composed of both large and small particles.

4. Tests made to determine the effect on average particle size of dust suspensions subjected to air filtration showed a marked decrease in the average particle diameter, the diameter of the particle with average surface area and the diameter of the particle with average volume. These data, together with the particle count arrestances, were used to calculate weight arrestances. These calculated arrestances checked reasonably well with the actual weight arrestance test values.

RELATIVE MERITS OF THE THREE FILTER TEST METHODS

From the standpoint of a practical test to be used in the rating of air filters each of the three methods discussed have certain inherent advantages and disadvantages. The more important of these may be summarized as follows:

Weight Method

The weight or crucible method of rating air filters has the advantages of high accuracy (estimated that results can be reproduced approximately within ± 1 per cent) and readily allows determinations to be made of filter life and dust holding capacity as well as arrestance. Furthermore, the performance characteristics can be determined when the filter is subjected to lint as well as dust. However, the weight method measures the ability of the filter to remove principally the large dust particles from the air. The smaller particles, which play a prominent part in arrestance determinations based on particle count and discoloration, have little effect on weight arrestances.

Particle Count Method

This filter test method in its present state of perfection requires too long a time to make a determination which can be considered statistically reliable. However, there is considerable evidence that arrestances based on particle count are better measures of the degree to which dust dangerous to human health is eliminated. Such an arrestance determination is not necessarily a measure of the elimination of nuisance dust, and the type and concentrations of dust normally experienced in other than industrial structures are probably not harmful. Therefore, it would appear that the most practical application of this test method would be its development to determine the degree to which dangerous dust concentrations are actually reduced in filtering installations.

Discoloration Method

Both the transmitted and reflected light discoloration test methods have a reasonably good accuracy (estimated at ± 3 per cent). Although these test methods have not in the past been conducted in conjunction with determinations of filter life, there is no reason why this performance characteristic cannot be determined as indicated earlier. However, it would probably be impossible to devise a lint test which could be used in conjunction with this method of determining arrestance.

This method is advanced as a means of determining the arrestance based on the reduction of discoloration or soiling power of the air-dust mixture and therefore probably best measures this property when reflected instead of transmitted light is used. Furthermore, when reflected light arrestance determinations are made it is not necessary to match the unused blank filter papers

as is the case with the transmitted light determinations, and the elimination of this procedure reduces appreciably the testing time. The amount of light reflected from the surface of the blank filter papers is practically constant, whereas the amount of light transmitted through these same papers usually varies considerably. Actually, both the reflected and transmitted light methods measure the reduction and discoloration only in relation to the test dust being used, and the arrestances vary greatly for the same filters with different test dusts.

For other than applications designed to eliminate dusts dangerous to health it would appear that the best filter arrestance or efficiency measure would be that which most accurately measured the percentage of nuisance dust eliminated. The question remains, however, as to whether the best measure of nuisance dust is the relative ability of the dust to discolor white surfaces or the weight of the large dust particles which are visible to the human eye. The photo-electric cell method is probably the best measure of the reduction in discoloration, while the crucible or weight method is the best measure of the reduction of the larger visible dust particles.

DISCUSSION

B. G. HEEBINK[12] (WRITTEN): This report emphatically indicates the need for a thorough, careful and very extensive study of air-borne dusts in all the possible conditions where filters might be installed. This should include lint and pollen as well as the granular or flocculent dusts. With this information, it may be possible to produce a synthetic dust (or possibly two or three dusts) which will duplicate the air-borne dusts.

It also indicates that a study of this nature will probably reveal that the particle size of air-borne dusts is very small. By using the weight method (which still seems to be the most practical) on these small particles, the efficiencies by weight will be dropped, and may be comparable to those by the discoloration and particle count methods.

The only method of test for particles of uniform size, such as, ragweed pollen grains, will undoubtedly be the particle count.

It is not clear to me why the special filter composed of two Filters *B* in tandem plus 14 layers of oiled cheese cloth had an arrestance of only 94 per cent by weight when one Filter *B* (according to Table 1) had an arrestance of 96 per cent on what I assume is the same dust, namely, 200 mesh Pocahontas ash.

R. P. WARREN[13] (WRITTEN): The fact that the weight test adopted in 1933 by the ASHVE was not suitable for rating filters for all purposes is undoubtedly true, and there has been a definite need for either additional standard tests or a single test which would do more than simply tell the weight of dust removed.

In considering comparative results obtained by the weight method and the particle count method, the writer suggests that another factor could be injected, namely, a comparison of the weight method and discoloration method with a dust count made in accordance with that method proposed in the Proposed New York State Code to be known as Industrial Ruling No. 10. This method uses the light field low power microscope count and samples are taken by means of a modified *Greenburg-Smith* impinger. The writer injects this thought because the many methods of dust counts give a variation quite as great as those reported by the different methods in this paper.

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[12] Research Products Corp., Madison, Wis.
[13] Buffalo Forge Co., Buffalo, N. Y.

In regard to the factors effecting the discoloration test results, the writer is particularly impressed with the findings that no appreciable variations were found in the results using different velocities through the sampling tubes. The writer seems to recall seeing a curve which plotted a correction factor against variance in velocity (this variance being that from air velocity in the duct sampled) wherein it was sometimes necessary to correct the readings as much as 15 per cent. This curve was either published in the *Journal of Industrial Hygiene and Toxicology* or in the *ASME Transactions*. This point is relatively important because if we can disregard the velocity in the sampling tube, that type of work will be somewhat simplified.

From the results published in Figs. 6, 7, 8 and 9 it is apparent that the three methods of rating filters are not comparable, and it is also apparent that each method measures a somewhat different characteristic of the filter. For this reason a published arrestance efficiency might be entirely misleading to a user of a filter. In other words, while the efficiency rating might be entirely true the filter might fall down and not give satisfactory performance.

Under that section of the paper relating to *Correlation of Filtering Theory with Test Results* the author states:

> If air filters were capable of eliminating both large and small dust particles with equal ease there would be little difference between the ratings obtained by the different methods. However, that such is not the case is indicated by Fig. 13 which shows the relationship between size of dust particle and filtering arrestance for three different filters.

As a continuance of this investigation the writer suggests that a typical filter be added to those already tested which exemplifies the filtering of dust from the air by electrostatic means. Such an investigation would indicate the variance of results using these three methods and at the same time show the comparative performance of the four filters already tested with a fifth filter which uses an entirely different principle of cleaning.

The writer has also been particularly interested in that part of the paper which discusses the possibility of computing probable results of the weight method from a particle count wherein the size of particle is classified. The application of this theory is not limited to the laboratory rating of filters, but might also be applied to the value of any filtering device in eliminating nuisances.

H. C. MURPHY (WRITTEN): This paper is a comprehensive study of various methods of rating air filters. It represents a logical continuation of the investigation started in 1926 and resulting in the Standard Code[14] which was adopted at the Semi-Annual Meeting of the Society in Detroit in 1933.

It is interesting to note that the conclusions of the present investigators coincide very closely with the opinions expressed by the Code Committee in 1933.

The present paper recommends the use of the weight method for rating air cleaners against nuisance dust—a problem in general ventilation work, and suggests the dust count method for air cleaning apparatus to create spaces free from health hazardous dusts.

The Code Committee reported to the Society in 1933:

". . . a code suitable for laboratory testing of air cleaning devices used in general ventilation work would be entirely unsuited for rating cyclones, dust separators, and the like, or for the selection and rating of devices intended to supply air for dust hazardous occupations in accordance with the standards of the U. S. Public Health Service. It therefore seems advisable to limit the use of this Code as defined under *Scope*, and it was also decided to recommend for this particular code the use of the weight method and non-proprietary instruments for determining cleaning efficiency."

Under *Scope* the report has the following:

"This Code is for the laboratory investigation and rating of devices used in general ventilation work for the sole purpose of removing solid impurities from the air. It

[14] ASHVE Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. (ASHVE TRANSACTIONS, Vol. 39, 1933, p. 225.)

is not intended for use in the rating of cyclones, dust separators or for devices used in the field of Industrial Hygiene. Ratings established under this Code are not to be confused with operating efficiencies in actual service—the dust concentration, the nature of the dust, the relative humidity and many other factors have a definite bearing upon the results obtained in actual service."

The present investigators find the weight or crucible method substantially as outlined in the present standard ASHVE Code to be the most accurate of the methods studied.

In view of the demonstrated accuracy of this Standard Code and the fact that it is widely and successfully used, it would seem that the action of the Council of the Society in announcing the withdrawal of the present Standard Code was precipitous and possibly based on misinformation.

The Society's present Standard Code is by no means perfect and it is not my intention to maintain that improvements in the test procedure have not been developed. However, it was and is the first and only complete test procedure which allowed a comparison of the various factors which go to make up the over-all efficiency of an air cleaner—that is, the dust arrestance efficiency, the dust-holding capacity and the resistance to air flow. A 1-in. pine board would be 100 per cent efficient in removing dust but no one would consider it an *efficient* air cleaner.

There have been complaints that the carbon factor in the *Test Dust* specified by the present code is unusually difficult to remove with any type of air cleaner. This is true but this component was selected in view of the fact that the test procedure is intended only as a *yard stick* and was not intended to be used to determine operating efficiencies in actual service.

It is not uncommon to see advertisements of 2-in. viscous unit filters under a guaranteed efficiency of 98 per cent to 99 per cent when tested in accordance with the ASHVE Standard Code *using practical dust.*

Of course it is difficult to say just what *practical dust* might be, obviously any cleaner would be 100 per cent efficient against *BB* shot or marbles, but unless a very difficult test dust is used, it is hard to express in percentage the difference in arrestance values of various cleaners.

M. H. KLIEFOTH (WRITTEN): It seems that the *photoelectric method* mentioned in Part 1 disregards entirely the factor of index of refraction, which seems important. Possibly D is meant to be a determination of density and index of refraction. But if D_1 and D_2 cancel out according to measurement, then the index of refraction is not considered. Otherwise D_1 and D_2 would not cancel out.

In Part 1 the importance of including the factor of refractive index measurement is brought out. This, however, means a particle count of the different indices of refraction upstream and downstream and would be rather ticklish for practical and even for scientific tests.

It is assumed that the photoelectric cell output was calibrated repeatedly during these tests and that proper corrections were made for the variations in the output. This writer has an aversion against photoelectric cells which is the result of years of experience with these devices with the final observation that photoelectric cells are not reliable unless they are calibrated continuously. The writer is interested to know what exact means were employed to measure the reflected light intensity.

In Part 2 the question of dirt or dust elimination or the selective efficiency of dust or dirt elimination hinges to a large extent on the question—is so-called *soiling dust* more objectionable or is the general dust which we find in average conditions. If we think only of soiling dust then we can adopt the photoelectric method but if we think of common dust as the important factor in life, then it seems obvious that the weight method is the proper one.

Page 42, Item 4—The writer wonders if the authors have considered that a mixture of Pocahontas ash and carbon dust will make aggregates of Pocahontas ash and carbon dust having entirely different dust wetting characteristics than the single

constituents. It appears that the monomolecular characteristics as well as the *p*H of these constituents affect the wetting and, therefore, the elimination of such particles.

In Part 2 under the heading *Continuous Discoloration Filter Test,* there seems to be an error or discrepancy. The first paragraph states that the dust feed was 20 g and at the end of the paragraph, we find this statement "when based upon a dust feed of only 15 g weight."

In the second paragraph under this same heading the author is not clear about the statement starting "although none of these tests were conducted . . ." and the conclusion that "continuous tests can be used. . . ." It seems that life and dust holding capacity are the same phenomena. If life can be determined, then dust holding capacity can be determined or vice versa.

Under the heading *Correlation of Filtering Theory with Test Results,* the paragraph starting *Air-borne dusts.* The calculated reductions seem rather theortical. We believe that a great many factors enter into this picture which will upset such theoretical considerations and these factors probably account for the exception "that discoloration arrestance calculated here is higher."

There seems to be a discrepancy between arrestance of a filter single and filter in tandem with 14 layers of cheese cloth backing. According to our way of reading, the single filter had an arrestance efficiency of 96 per cent while the filter in tandem with 14 layers of cheese cloth backing had only 94 per cent. What is the explanation? Should these tests not be compared? Were the dusts different? Was the test method changed?

Page 1—assuming spherical particles—We would be interested in knowing the exact method used for particle size measurement.

(a) Particles are not spherical but have three dimensions.

(b) Particle count has to be weighted. By this, we mean particles of approximately the same volume fall into one class but particles of a different volume fall into another class. Therefore, particle size and particle volume distribution seem important.

The writer does not believe that the average diameter micron determination is of any great importance but rather thinks that the particle volume distribution and the specific gravity of the particles play an important part.

The writer agrees fully with the authors' conclusions—"Relative Merits of the Three Filter Test Methods." It is believed that filter efficiencies should be expressed as a result of three to four different dust combinations rather than being presented on one dust only. This, in the writer's opinion, will bring the most practical present day solution.

ARTHUR NUTTING (WRITTEN): This paper is timely because there is a trend today to specify air filter characteristics in terms of performance under some particular test method. During the past few years, the number of test methods developed has exceeded the number of new kinds of filters developed. This is evidence of a great public and industrial interest in the subject of air cleaning. Certain industries have developed their own methods which intended to measure performance in terms that can be interpreted to suit their own unique requirements. In general, however, the ASHVE weight method and the Bureau of Standards discoloration method, or some form of it, are most widely accepted.

The discoloration method is usually employed to test air filters which have a high order of arrestance efficiency. Performance cannot be differentiated by the weight method when the arrestance efficiency approaches 100 per cent. In this case, the discoloration method brings out whatever difference exists between filters of this class.

In my opinion, the most valuable characteristic of the discoloration method is that it enables one to measure filter arrestance efficiency on atmospheric impurities, and therefore lends itself to filter arrestance efficiency measurements in the field. There has long been a real need for a simple and practical field test procedure, and

I believe the discoloration method fills this need for air cleaning devices having a high order of dust removal. I like to think of this method as being one strictly reserved for the measurement of atmospheric impurities and not to be associated with artificial testing methods.

Professors Rowley and Jordan made discoloration tests with atmospheric dust and certain artificial dusts. It is quite probable that subsequent tests would prove the inadvisability of using artificial dusts. A year or so ago, Prof. Rowley made certain tests by the Weight Method which show how the nature of the dust affects weight arrestance efficiency. In particular, he found great differences in his results when he used different kinds of carbon black and lampblack—and he even found differences with two samples of lampblack of the same grade and made by the same manufacturer but made at different times. Now if dusts, which for practical purposes appear identical, can cause measurable weight efficiency differences, they will cause far greater differences when the discoloration method is used.

At Louisville, for the past two years, all our tests on air filters have been made with both these methods employed simultaneously. Our discoloration tests have been made with both reflected and transmitted light. We obtain, however, identical results with the light reflected and with it transmitted through the dust spot.

In order to accomplish this, one must be careful when making dust samples for reflected light tests, that the dust spot is not too dense. We have found that if the dust density on the target is controlled so that it will absorb no more than 25 per cent of the light, the amount of light absorbed will be proportional to the quantity of dust on the target. If the dust is allowed to build up to a greater density than this, the straight line proportionality is destroyed. Referring to Fig. 5 of this report, it will be observed that the curves, when carbon black was used, could be straight lines down to the point where 45-50 per cent of the light was reflected. The advantage of using reflected light in discoloration tests in that dust deposits of unequal density can readily be compared. Equal volumes of air can be sampled from each side of the cleaning device and the resultant spots compared photometrically. Field tests are thus greatly simplified for we eliminate the necessity of obtaining deposits of equal density which often becomes an awkward procedure in the field.

C.-E. A. WINSLOW (WRITTEN): This paper constitutes a most important contribution to the complex problem of filter testing. It should be of material value to the Society in the formulation of a new code for filter testing which is so eminently desirable.

This study shows very clearly that there can be no single method applicable to the testing of filters under all possible conditions. At one extreme, the weight method, which reveals the total amount of foreign matter carried by the air, seems entirely appropriate for determining the removal of the large particles, which are important from an aesthetic standpoint. At the other extreme, the particle count method is the only procedure calculated to determine the efficiency of a filter in removing the very minute particles, which are those of primary importance in dealing with dusts detrimental to health. The large particles which chiefly determine the results obtained by weight determination are of no importance from the health standpoint since they are taken out in the upper parts of the respiratory tract; while it is the particles below 1 μ (micron in size) which penetrate to the lungs and are active in producing silicosis and other pulmonary diseases.

It seems clear that in formulating a new code, alternative methods will have to be recognized, depending on the major objective of the dust removal process in a particular case.

THEODORE HATCH[15] (WRITTEN): Atmospheric dust varies in composition, particle-size and concentration from one locality to another and from time to time in a

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given locality, notably with changes in the season. In composition it is not homogeneous; on the contrary, one constituent may be relatively coarse in size and another concentrated in the finest fraction. It contains crystalline and amorphous mineral matter, inert and vital organic material, individual particles and aggregates, dry and oily particles and even liquid droplets, light-colored and dark particles, etc. The undesirable effects resulting from particulate contamination of the air supplied to an enclosed space are well known and require no special mention. It should be emphasized, however, that certain of the damaging conditions are produced not by the dust as a whole but by its different constituents or by different size-fractions of the complex suspension. Thus, the visible *dust* on furniture represents the coarse fraction and aggregates which settle out rapidly. Curtains and other fabrics are soiled by the finer oily and adhesive constituents, usually dark in color. Delicate electrical contacts are interrupted by the collection of non-conducting particles on the contact points. Asthma will be produced in certain individuals by relatively coarse plant pollens and in others perhaps by extremely fine organic constituents. It follows that there is no single criterion for measuring the dirtiness of air or the efficiency of an air-cleaning device, but rather, one must select a testing procedure which is designed to meet the requirements of the particular problem. The specifications for a hospital, for example, may differ from those for a department store or for a delicate manufacturing process; and the requirements for a community using hard coal predominately will not be the same as for one burning soft coal or oil.

The authors have investigated the characteristics of three test methods for measuring filter efficiency and have obtained widely different results, which, as they point out, would be expected from the fact that the same scale of measurement is not employed in the three procedures. One test favors coarse particles, another the finest fraction of the dust, while the third falls in between. Thus, the three methods cannot be compared when the particle-size distribution and optical properties of the different fractions of the test suspension are not known.

In view of the foregoing it seems to be premature to discuss the relative merits of the various methods of measuring the efficiency of air-cleaning devices until more precise specifications of air cleanliness have been developed; that is, until the objectives are more clearly defined. When this has been done it may be found necessary to employ different criteria and test methods to evaluate the performance of filters and other cleaning devices under different situations. In the meantime, the use of weight efficiency alone may properly be criticized by those who are concerned primarily with the damaging effect of fine particles. The measurement of efficiency by particle count or discoloration, on the other hand, will unjustly penalize filters which are required simply to remove the rapidly settling constituents of the atmospheric dust.

My comments are not in criticism of the authors' investigation, which has brought out much valuable information. In a sense, however, the study reveals the manner in which filters perform against dusts of different sizes to a greater extent than it does the relative value of the different test methods. In this connection one regrets that a more sensitive dust counting and size-measuring technique was not employed, as it would have shown even more strikingly the importance of particle size. By means of the generally employed dust sampling instruments, the average particle size of dust in normal air is found to be approximately 1 μ and not 7-9 μ, as reported in this study.

LEONARD GREENBURG, M.D. (WRITTEN): It seems to me that the paper should include a statistical description of the particle size distribution of the dusts.

The largest amount of the dust appears to be composed of large sized particles. In this respect, if these filters are to be used for ordinary atmospheric dust, the experimental tests are not realistic since they utilize a test dust of a size so much greater than that found in ordinary atmospheres. In this connection it is interest-

ing to note (Fig. 13) that the lower limit of effectiveness of the filters is approximately 10 μ.

For the most part, the test data are set forth in Table 1, but there is nothing in this table or elsewhere to indicate the number of tests made in each case and the variation found between the test, for example, showing lowest efficiency and highest efficiency.

Throughout all of this study there seems to be a lack of data with reference to the results of particle count methods of determination.

PROFESSOR ROWLEY: A statistical description of the particle size distribution of these dusts has been given in the articles referred under footnotes 1, 2 and 3 of this paper, which answers Dr. Greenburg's first comment.

The test dusts used in these experiments were those generally accepted and used by the filter manufacturers and were similar in particle size to the Standard ASHVE Code filter test dust. The statistical sizes of these dusts are obviously larger than those for normal air-borne dust, and it was for this specific reason that tests were made and reported using normal air-borne dust in conjunction with the particle count and discoloration methods of test.

Answering Dr. Greenburg's third comment, in every case at least one, and in most cases more, check tests were made in determining the values reported in Table 1. The degree of variation found between check tests for the weight and discoloration methods has been indicated in the paper under the heading *Relative Merits of the Three Filter Test Methods.* It is here stated that it is estimated that the results obtained by the weight method can be reproduced approximaely within $\pm$ 1 per cent, and that the discoloration test results can be reproduced approximately within $\pm$ 3 per cent. These ranges indicate actual ranges in percentage units and were based on observations of the variations found on consecutive tests of the same or identical filters using the same test method and the same test dust. In some cases as many as 5 to 10 check tests were performed with the results all indicating the discrepancies previously stated.

The amount of test data reported for all three test methods was approximately the same with the exception of the results for the continuous tests performed by the weight and discoloration methods. Because of the volumnity of the original data it was deemed inadvisable to report more than a summary of these results. With reference to the number of tests performed in making the settling chamber and dust counter particle count arrestance determinations it was stated previously that for each determination made, five samples were taken from each side of the filter and counted and averaged, and that the arrestance results reported in Table 1 are the average of five separate determinations or tests. Thus, in making one determination as reported in Table 1 it was necessary to make 50 dust counts in order to determine a statistically accurate average determination. The possibility of duplication of this determination upon making another 50 dust counts may probably be placed roughly at $\pm$ 5 percentage units. It was for these reasons that it was stated that this test method in its present state of perfection requires too long a time to make a determination. However, it must be realized that arrestance determinations based on the particle count method are probably the best measures of the degree to which dusts dangerous to human health are eliminated. Therefore, it would appear that the most practical application of this test method would be its development to determine the degree to which dangerous dust concentrations are actually reduced in actual filtering installations.

PROF. F. B. ROWLEY: It is impossible to answer in detail all this discussion presented here. It does, however, indicate that there is a dissatisfaction with our present method of rating air filters. It is also evident that a field method of testing is desirable. The photoelectric cell calibration referred to by Mr. Kliefoth is not

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necessary, since the tests are comparative and the cell is virtually calibrated for each test.

It seems to me that Mr. Murphy's discussion sums up the state of affairs when he suggests that tests by the Code method should not be confused with the results from practical application of filters. I believe that there is a general feeling that the Code does not give results which represent those that may be expected from the filters in service. To this extent the Code should certainly have consideration and revision. The whole question needs careful study, and it is doubtful whether or not a Code can now be devised which will meet all criticisms.

No. 1170

COMFORT IN HIGH ALTITUDE FLYING

By D. W. Tomlinson,* Kansas City, Mo.

DURING the past 10 years the improvement that has been made in the airplane as a whole is almost phenomenal. The heating and ventilating system of passenger planes has always presented a major engineering problem, and the problem has not yet been completely solved. However, all air travelers will agree that the safety and comfort provided for the present day airline passenger has been greatly improved, and the contrast between the heating and ventilation of the modern Stratoliner of today and the aircraft of 10 years ago shows that some notable advances have been made.

Ten years ago heating and ventilating was considered of more or less secondary importance. The supply of cold air was taken in through small *horns* located at each seat, these horns extending through the fuselage to the outside. They were designed so they could be turned to take in the desired amount of air or turned up or back to stop the flow of air. Naturally, considerable noise, vibration, and quite often fumes from the engines were brought into the cabin, but at that time it was considered by most people as just part of air travel. No spent air or ventilation valve was provided to help control the amount of foul or used air escaping from the cabins. Leaks around doors and windows, which were usually plentiful, were depended upon to solve this problem.

The conventional heating system in airplanes of 10 years ago usually consisted of a metal shroud covering the exhaust tail pipe as shown in Fig. 1. This tail pipe was ordinarily on the under side of the plane and quite often extended almost the full length of the cabin in order to obtain sufficient heat. The air entered the duct thus formed at the forward end between the exhaust pipe and the shroud and after passing along the hot exhaust pipe, was admitted to the cabin through adjustable sliding valves in the aisle between the seats, or through tubular ducts along each wall near the floor as indicated in Fig. 1.

Admittedly poor circulation, and quite often inadequate heat, were obtained with this arrangement, yet it was many years before it was changed. The admittance to the cabin of exhaust gases and carbon monoxide from breaking exhaust pipes and loose connections was always feared, and now it is sometimes wondered how the airlines operated as long as they did without more

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Presented at the 47th Annual Meeting of the American Society of Heating and Ventilating Engineers, Kansas City, Mo., January, 1941.

serious trouble from the hazard of asphyxiation and fire in the cabin heating systems of that period.

In 1933, when the Douglas DC-1 was built for TWA, the first cabin steam heating system, to the author's knowledge, was installed for airline use. The system was of the low-pressure type, operating under only a few pounds pressure. The water entered the boiler at the bottom by gravity feed and went out the top as steam to the steam radiator, which was located in the fuselage below the cabin floor. The boiler was located in the exhaust manifold in one engine nacelle which is the streamlined compartment housing the engine on the wing. Cold air picked up at the nose of the plane was directed by an electrically operated automatic temperature control either through the steam radiator or around it, depending upon the cabin heat demand. Fig. 2 shows the heating and ventilating system of the DC-2, which is identical with that of the DC-1. After this first DC-1 steam heating system,

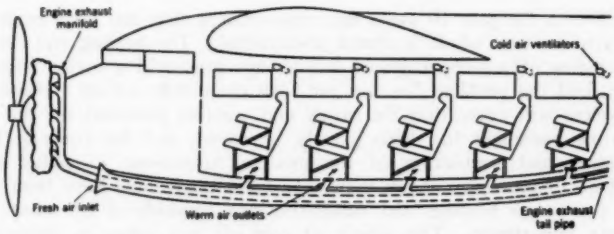

FIG. 1. TYPICAL HEATING SYSTEM OF 10 YEARS AGO

fast strides were made to the present-day heating and ventilating system now installed in the Stratoliner, designed and built by Boeing.

The heating and ventilating system of the Stratoliner consists actually of five separate, though intimately related, systems, as follows: (1) Fresh Air System, (2) Spent Air System, (3) Warm Air System, (4) Steam System, and (5) Ground Air Conditioning System.

Fresh air is supplied to the cabin by means of an air scoop on the upper side of the fuselage through which air is forced by the forward speed of the airplane into a distribution duct which extends the full length of the passenger cabin and which is located on the ceiling as indicated in Fig. 3. This is divided longitudinally into two sections; the lower of which contains grilles through which air is distributed directly into the cabin while the upper section is a supply duct for the individual outlets which are located at each seat and upper berth. The individual outlets are also provided in the ladies' and men's dressing rooms and other sections of the plane forward of the passenger cabin.

After the air enters the scoop, it passes through a centrifugal water separator device which is an integral part of the duct just inside the fuselage skin, and which is intended to prevent water from being forced into the distribution duct in the cabin. There are two valves located in the outlet duct behind the water separator, the foremost one being used for closing the air inlet during the time the cabin is supercharged or during extremely cold weather, when

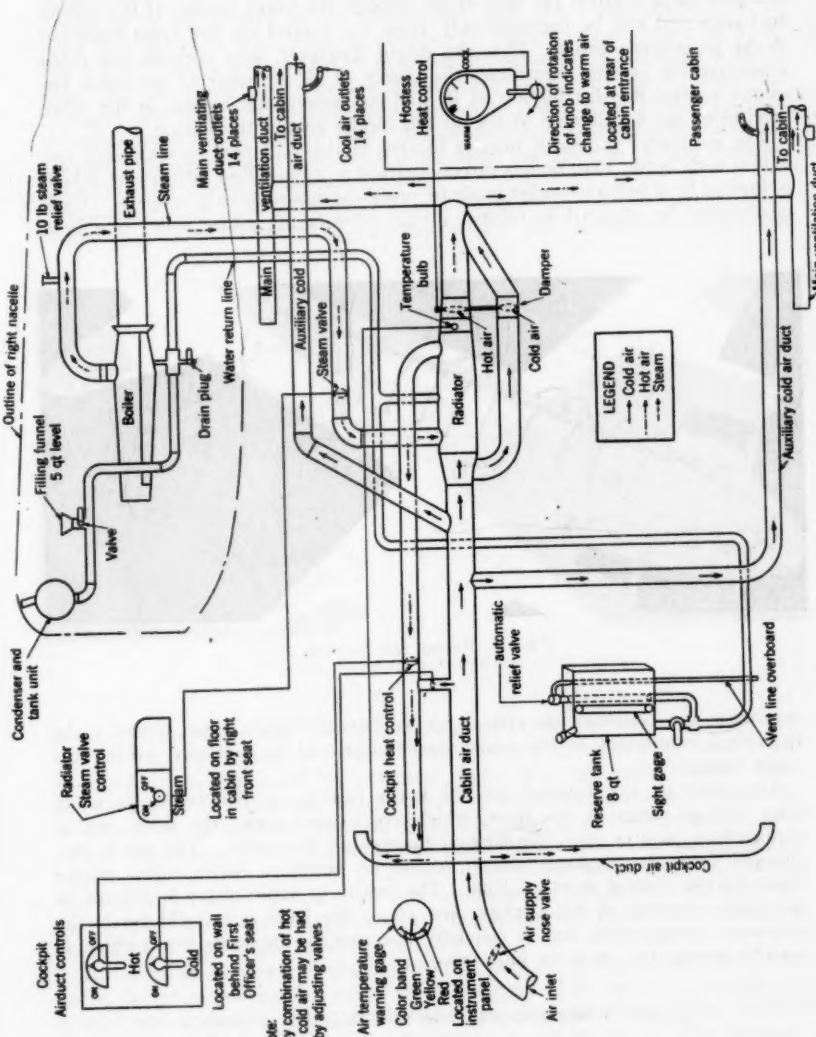

FIG. 2. HEATING AND VENTILATING SYSTEM OF DC-2 PLANE

the outside cold air is not necessary. It is operated from the cockpit by means of a lever on the floor behind the seat occupied by the First Officer. The rearmost valve controls the flow of air through the lower section of the ceiling duct only and can be operated only from the control on the front bulkhead in the passenger's cabin. Thus the Flight Engineer, who controls the cabin supercharging system, determines the source of all air entering the cabin but cannot control the distribution of it; and conversely, the hostess in the cabin determines the distribution of the air but cannot control the source.

The individual ventilator nozzles located in the wall next to each seat and within each berth, have both a directional and a volume flow adjustment which is formed by a ball-and-socket made of sound-deadening plastic material. Each nozzle may be adjusted to release a high velocity air jet in order to obtain

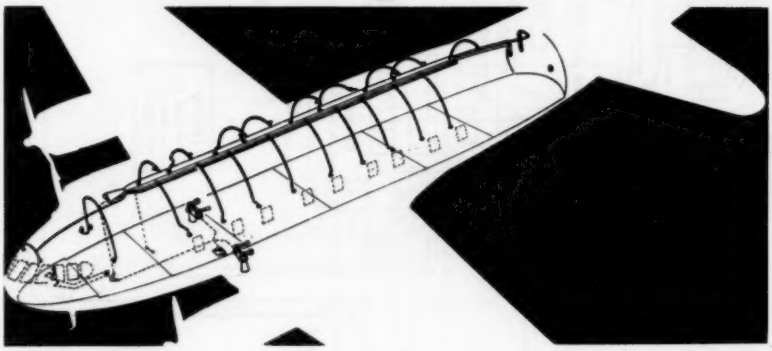

FIG. 3. FRESH AIR SYSTEM

maximum air recirculation effect. A rubberized fabric hose connects the individual ventilators to the main supply duct and is concealed behind the cabin lining.

The spent air is exhausted into the lower fuselage section below the cabin floor through vents in the floor, which are located under the seats and so arranged as to give even distribution throughout the cabin. The air is discharged from the airplane through either an auxiliary outlet valve, or the supercharger control units or both. The auxiliary outlet valve is located in the lower covering of the airplane just aft of the front spar bulkhead in the accessory compartment and is manually operated by the same lever which is used to operate the fresh air inlet valve.

CABIN SUPERCHARGING AND THE WARM AIR SUPPLY

The pressure, air supply, control, and temperature conditioning system is provided in duplicate, there being one system on each side of the airplane. (Fig. 4) either system being capable of maintaining satisfactory cabin temperature and pressure should the other system fail.

All doors and hatches open inward and, therefore, are sealed against rubber gaskets under the influence of cabin pressure. The skin or fuselage covering continuity is not broken at the intersection with the center section of the wing and forms a cell which is circular in section, cigar-shaped in form. There are 17 passenger compartment windows measuring 12 in. high x 16 in. wide, and the panes of which are a single-ply section of 5/16 in. Plexiglas. Plexiglas was chosen for its high strength-weight ratio. The cabin is designed to withstand 6 lb per square inch internal pressure, although the maximum internal working pressure intended for transport operation at present optimum altitudes is but 2.5 lb per square inch.

The cabin floor level is located below the center line of the fuselage at its largest section. The cargo compartments and the accessory compartment

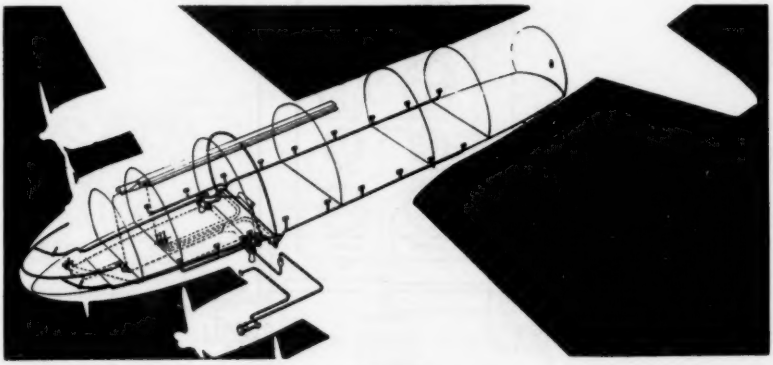

FIG. 4. HOT AIR AND SUPERCHARGING SYSTEMS

between them under the floor are maintained under full cabin pressure and are part of the circulatory system. The fuselage is pressurized only to the rear end of the cabin and at that point, there is a removable circular hatch in one side of the hemispherical bulkhead to enable inspection of the interior of the tail of the airplane. An emergency air valve set to relieve at a pressure of 2.65 lb per square inch is also located in the bulkhead as a safeguard against building up excessive internal pressure. This valve has a capacity equaling the total overloaded discharge of both cabin superchargers at sea level and during maximum supercharger speed, without causing the cabin pressure to exceed 3.6 lb per square inch.

This airplane carries a Flight Engineer as part of the flight crew, one of his duties being to handle the cabin supercharging. His station is behind the co-pilot and he has an instrument board in the side wall of the flight compartment on the right hand side of the cockpit as shown in Fig. 5. The pressure-cabin instruments are mounted on this panel in a group by themselves, separated from the surrounding instruments. In this group and at the lower left is a supercharger discharge pressure gage and a two-way selector valve for connecting the gage to the right or to the left supercharger in order that the

pressure rise may be known. On the right is a suction gage with a three-way selector valve connected to it. When the latter valve is in the central position, the suction gage indicates cabin pressure referred to outside pressure. When the valve is in the left or right position, the suction gage indicates depression in the throat of the flow-measuring venturi of the right or of the left pressure air supply system. A selector valve is located in the uppermost portion of the panel for the purpose of manually selecting operation on either the right or left outlet valves, or both as desired. Below this space, a vertical speed indicator, referred to cabin pressure instead of to atmospheric pressure, indicates the rate of cabin pressure change in terms of feet per minute. Next to this gage

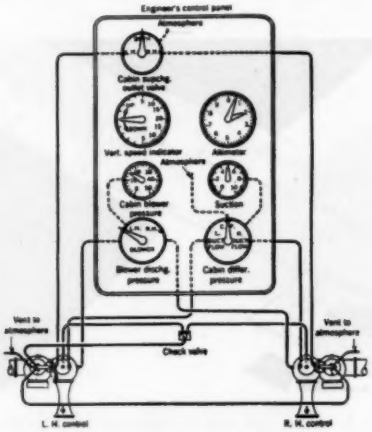

FIG. 5. INSTRUMENT AND SUPERCHARGER CONTROL UNIT CONNECTION DIAGRAM

is an apparent altitude green warning light to inform the engineer of the absolute pressure in the cabin. The green light can be pre-set by a knob below it to shine at consecutive apparent pressure altitudes of 8000, 15,000 and 18,000 ft.

The air to be compressed is drawn from an inlet in the leading edge of the wing adjacent to the wing fillet. From there it passes through a water separator, through the supercharger (Fig. 6) which is driven by the engine, through the steam radiator in the leading edge of the wing, through the intercooler to reduce the temperature of the air, through the inflow control regulator, on through the cabin, out of the cabin through the floor into the compartments under the floor and then through the outflow valve which is incorporated in the unit containing the inflow valve.

If the cabin temperature is too cold, feedwater pumps supply water to the boilers and these in turn supply steam to the radiators, thus raising the temperature of the incoming air passing through the radiator. If the cabin temperature is too high, the feedwater pumps do not operate, consequently

the air passes through the cold radiator and the intercooler. On warm days during supercharged operation, the supercharger discharge temperature is such that it is necessary to operate with the intercooler valves open for maximum cooling.

The inflow of pressure air is fixed by the automatic controls at an impact value corresponding to the impact produced by 250 cfm per blower at sea level. Except for changes due to cabin leakage, the flow value is unaffected by changes of other conditions, such as blower speed. The absolute pressure in the cabin is maintained substantially constant between 8000 ft and 15,000 ft pressure altitude. At altitudes higher than 15,000 ft, the differential pressure of the cabin referred to the atmosphere is automatically held at approximately

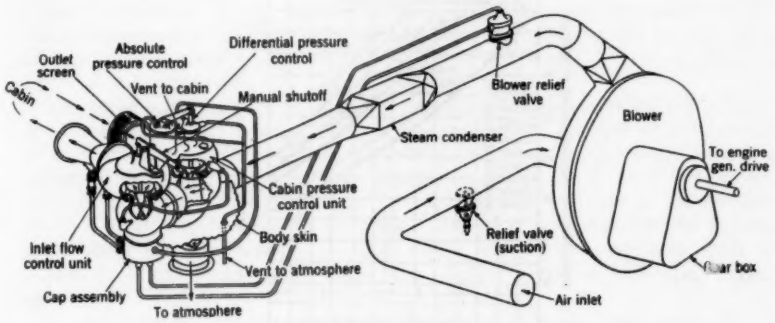

FIG. 6. CABIN SUPERCHARGING CONTROL INSTRUMENT

2½ lb per square inch, resulting in an apparent cabin altitude of 11,000 ft at an actual altitude of 19,000 ft (see chart Fig. 7).

The pressure air flows into the spherical casting of the cabin supercharger control and then rises past an inlet flow control valve and flows horizontally into a measuring venturi to be distributed to the cabin. This inlet valve, incidentally, acts as an outflow check valve if the air flow ceases. It may also act as an inflow relief valve if the cabin pressure should tend to fall below the atmospheric pressure on the outside of the cabin.

Air to be exhausted from the pressurized area passes through a screen and then escapes to the atmosphere past an outlet valve which is also a part of the supercharger control unit. The seat of the outlet valve is extended downward through the inlet duct, forming a diffuser; in this manner the seat and diffuser of the outlet valve are heated by the incoming air from the superchargers to prevent formation of ice which might form under the cooling influence of the expanding outflowing air. The outlet valve also acts as an inflow relief if the cabin pressure tends to fall below the atmospheric pressure on the outside of the cabin.

The inlet valve and its motor mechanism are identical in construction to that of the outlet valve, both being of the tulip poppet type, although each motor mechanism is regulated by different control devices. The valve stems

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are directly attached to reciprocating air motor pistons. The valves are spun from stainless steel sheet metal and the pistons from magnesium, which is also used in the control unit castings. The stainless steel valve stems are guided by means of self-lubricating bronze bushings which space the pistons from the cylinder walls with a small clearance.

Regulation of the inlet valve position is accomplished through adjustment of the intensity of action of the pressure air upon the motor piston. The regulating mechanism is supposed to hold the poppet valve to within about one-thousandth of an inch of the position required by this sensitive mechanism. Since the inlet valve motor is affected by the depression in the flow-measuring venturi, the cabin supercharger serves merely as a ventilating fan when the

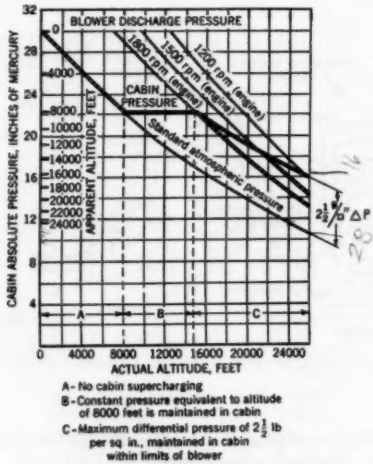

FIG. 7. CABIN PRESSURE CHART

cabin is not being supercharged. However, when the auxiliary or fresh air ventilation inlet and the auxiliary outlet are closed, the inlet valve motor becomes partially responsive to the position of the outflow valve so that there may be some flow compensation in proportion to the magnitude of cabin leakage as indicated before the inlet valve is designed to serve as an outflow check if the pressure in the supply system falls below that of the cabin.

The air motor of the exhaust or outflow valve is affected by an evacuated bellows which is sensitive to absolute pressure in the cabin and to a piston which is responsive to the differential pressure between the cabin and the atmosphere; hence this is the cabin pressure controlling medium.

During hot weather it is preferable to divert the incoming air from the superchargers into the fresh air ventilation duct in the cabin ceiling. Thus, the entire supply of air is thereby made available at the overhead grilles and individual ventilators, from which it provides the most cooling effect. The second optional flow course consists of a series of ventilation grilles in the

side of the cabin near the floor to provide better warm air distribution during cold weather. During average weather the air may be directed to both the overhead and to the floor duct systems or either one. If necessary, the major portion of the heated air can be forced into nozzles extending along the lower edge of the pilot's compartment windshield to defrost the windshield.

There is a cross-over duct between the right hand and left hand systems to insure the even distribution of air throughout the cabin in the event that an inboard engine or supercharger is stopped. Each of the two supercharging systems is capable of providing sufficient heated air to warm the cabin during the most extreme weather conditions and is capable of supplying a quantity of air well in excess of the requirements of the passengers. It is further able to maintain the normal cabin pressure events even at the maximum operating altitudes of the aircraft. The pressure control unit prevents the superchargers from overloading during normal operation regardless of the speed of the inboard engines; however, if one engine stops, the supercharger on the other engine is automatically allowed to overload if additional flow is required to maintain cabin pressure.

The cabin superchargers (Fig. 6) are centrifugal compressors having integral speed-increasing gear boxes. There is one mounted on the rear of the firewall of each inboard nacelle and they are each driven by a flexible shaft from the generator drive of the engine. The superchargers are G. E. Type B-1, which is rated at 225 cfm at an absolute pressure of 9.35 PSI and a temperature of 70 F at an engine speed of 1500 rpm. The gear ratio between the flexible drive shaft and the impeller shaft is 7.75:1 and as the generator shaft is geared to 11.625 times engine speed, the impeller speed at 1500 rpm engine speed is 17,500 rpm.

A characteristic of all centrifugal blowers is that for any given set of conditions there is a minimum flow required for stable operations. It would, therefore, be impractical to throttle the discharge flow below this critical value without providing some means of relieving the discharge duct pressure in order to prevent surging. A blower relief or surge valve was designed to serve this purpose and is located in the discharge duct just behind the blower. The surge valve is responsive to flow or dynamic pressure at a chosen location in the system, also a certain static pressure in the blower discharge duct is necessary to cause opening of the valve. Without the action of the flow controls, the valve would open as soon as the static pressure in the duct became great enough to lift the piston valve against its weight, as the upper side of the valve would then be exposed to atmospheric pressure. Such action would, of course, be undesirable because high duct pressure is essential to normal operation, regardless of flow. The diaphragm attached to the sliding valve in the cylinder head is subjected on the lower side to cabin pressure and on the upper side to inlet venturi suction. A suitably large venturi depression, that is, sufficient flow for stable operation, will force the diaphragm and valve upward against the downward action of the spring and thus close the vent from the upper side of the piston valve to the atmosphere. Air then leaks through a small orifice at the center of the piston valve and both its upper and lower surfaces are exposed to duct pressure. The valve is thus held closed because the effective area above it is larger than that below it.

An inward opening valve is located in the supercharger inlet duct within the wheel-well of each inboard nacelle. This is designed to open at a suction

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pressure of ½ lb per square inch within the inlet duct and provide a source of air for the blower in the event that the opening at the leading edge of the wing is closed by ice or snow. The Boeing airplane has four engine nacelles and the two nacelles located nearest the fuselage are called the inboard nacelles.

An automatic temperature control, which consists of an invar rod, is located in a position in the cabin to receive a sample of the air and average cabin temperature as the outlet air passes through the louvers in the cabin floor. This invar rod, through expansion and contraction, operates a micro-electric switch; this in turn puts into action an electric solenoid which operates a valve connected to the plane's hydraulic system. A hydraulic pump in turn operates the feedwater pump that supplies water to the boiler.

The regulation of heat output is accomplished through thermostatic control of the speed of the feedwater pump.

The water supply for each of the duplicate heating systems is about one pint, and is contained in a sump located below the exhaust pipe of each inboard engine. This position is necessarily the lowest point in the system so that all of the water will drain back to the sump when the engines are shut down. Due to the conical shape of the sump, freezing of the contents will not damage it. When the engines are started, heat from the exhaust gases is transmitted to the sump and produces sufficient heat to melt any ice and enough pressure within to force the water out into the circulating system. As the steam rises through the tubes in the system, trapped air is compressed and increases the pump pressure. An airdome, located on the feedwater pump, provides a means of relieving this pressure by acting as an expanding chamber. Excessive pressures are controlled through a pressure switch which automatically cuts out the solenoid circuit to the pump; the circulation in the system then stops until the pressure falls off.

The boiler is a finned cylinder of heat resistant alloy, there being two inserted in the exhaust pipe of each inboard engine to extract heat from the exhaust gases. A threaded spindle is located within the bore of the cylinder. The feedwater pump forces the water along the thread and into contact with the inner surface of the bore at very high velocity, forming steam. The centrifugal force separates any water from the steam being formed, keeping it against the hot surface. The use of fins on the exterior of the boilers provides large heat uptake areas in a small space. As a consequence, there is a very high heat transfer rate although the boiler is very light and compact. Each of the individual heating units has two boilers, each boiler being rated at 40,000 Btu.

The steam generated by the boiler passes through the sump and is admitted to the upper header of the condenser, or radiator, core and distributed from the header to ten banks of corrosion resistant tubes. The heating of the air is naturally accompanied by condensation of the steam, and this condensate is drawn from the bottom header of the condenser by the feedwater pump. The described condenser is highly efficient and yet rugged. It is pressure tested at 1000 lb per square inch, although the maximum steam pressure is limited by the automatic heat controls to 125 lb per square inch.

The ground air conditioning equipment consists of a self-contained unit of the direct expansion type, and is capable of furnishing approximately 55 F. Attachment fittings for the ground air conditioning apparatus are provided on the lower right side of the airplane just behind the wing fillet trailing edge

and are covered by a hinged door. A duct located under the floor, connects with another vertical duct in the rear of the cabin, which serves to connect the ground air conditioning duct with the fresh air distribution duct on the ceiling. Conditioning air is thereby admitted to the cabin through these special ducts to the fresh air system ducts, and from there is exhausted into the cabin either through grilles in the ceiling duct, or the individual ventilators at each seat, or both.

In conclusion it will be readily apparent that great progress has been made to improve the comfort of airplane passengers, and the development of the special type of heating and ventilating system for Boeing Stratoliners marks another milestone in the progress of providing comfort for high altitude flying.

DISCUSSION

E. O. EASTWOOD (WRITTEN): Members of the Society have not made the contributions that may have been expected of them, to the heating and ventilation of airplanes.

The development of such systems applicable to airplanes has been due to efforts of engineers outside the profession of heating and ventilation. Several reasons are accountable for this. The demand has been small and gradual. Most airplane work, if not secretive, represents the manufacturer's investment in research and experimentation, and new designs have been guarded as long as possible.

The history of development has been strikingly repetitive. First hot air, then steam and water with automatic controls. Mr. Tomlinson's valuable paper will be better appreciated by those who realize the limitations imposed on aircraft installations. Space and weight are the most important limitations, and before the construction of a new airplane is undertaken a mock-up is built. Practically every item entering construction and equipment is duplicated in this imitation model, the cost being comparable to the cost of a finished plane. The principal function of a plane is to carry a load. In passenger service a pay load amounts to approximately 10 per cent of the total weight. In fighting craft the disposable load is the important factor. One pound of weight saved in construction is considered worth $100.00.

Fortunately, freedom from dust and the satisfactory humidity conditions encountered simplify the engineers' problem. Control instruments subject to influence of vibrations may not be used, and manually operated instruments over-ride automatic controls. This paper describes the hot air system using engine exhaust shroud. The author refers to the later type of steam heating installed on the DC-1, the hot-water type installed on the DC-2, an indirect hot air-hot water system and finally the hot air-steam system used on the Boeing Stratoliners, requiring supercharging of cabins for high altitude flying. The hot air system is in use on the B314, Boeing, 74 passenger Clipper where two of the 4, 1500 hp engine exhausts are equipped with stoves. Air is changed every 3 min. Either engine can supply all of the heat required—estimated to be 400,000 Btu per hour. The Clipper weighs 42 tons—the average weight of a railway car loaded with coal. High altitude flying is desirable on account of weather conditions, but requires provision of comfort for the passengers and structural problems resulting. At 20,000 ft the atmospheric pressure is approximately 7 lb per square inch, temperature —12 F, but a much lower temperature may be experienced.

There are no published data concerning the coefficient of heat transmission for a thin metal wall, lightly insulated and subjected to a wind velocity of 250 mph, at a temperature of —40 F. Radiant heat effect must interfere with passenger comfort and testimony strengthens this conclusion even when the ambient air is maintained at 70 F. As an example of success in saving weight, the two hot-water

radiators in use on the DC-4 weigh each 36 lb. Each has 1 sq ft of surface and liberates 400,000 Btu per hour. Boilers weigh 45 lb each. A complete circuit of the heating medium is made in one minute.

Others, which are special considerations, in airplane design and installations are noise, and the short life of parts of heating systems under prevailing conditions.

W. A. RUSSELL (WRITTEN): This paper is of special interest to me. It recalls a statement made by Professor Carhart lecturing to our physics group at the University of Michigan. It was a few years after the successful flights of the Wright Brothers at Kitty Hawk and lest we be carried away with too much enthusiasm over the possibilities of aviation, he cautioned us never to expect practical commercial results from a heavier than air flying machine. We see undreamed of success today in aviation and marvel at dual units each providing a five-way system of heating and ventilating for the Stratoliner, supplying that additional comfort that attracts ever increasing passenger travel.

Let us go back to the Twenties when we speeded around the Southwest in corrugated metal tri-motored planes of a well-known make, which through their safety record gained public acceptance and support for air travel in the Southwest and in America. The heating plants in later ships were effective and while gas leaks could occur between the shroud and hot exhaust pipe from which the warm air was brought through sliding valves in the aisles between the seats, I have no recollection of discomfort from such leaks. The fresh-air advocate, who invariably travelled in the seat ahead, persisted in twisting the small horn ventilator to a wide open position, resulting in noise and rapid cooling of seat occupants immediately behind.

Then came Mr. Tomlinson's conception of travel comfort built into the Douglas DC-1 for TWA; the first cabin steam heating system. There was no precedent to follow, no accepted method of controls, and the departure from heated exhaust pipe procedure, a relic of automotive practice, opened up an entirely new field. There was an unprecedented job to be done and the aeronautical engineer accomplished it in masterful fashion even to providing cooling for the latest type planes while grounded, an important hot-weather service.

One of the duties of the Flight Engineer is to handle cabin supercharging, providing 2.5 lb internal working pressure per square inch and all the other cabin conditions for passenger comfort which are constantly reflected on his instrument and supercharger control unit. He controls the source and condition of all air entering the cabin but the hostess determines the distribution of the air and cannot control the source. Even these interrelated duties may some day be accomplished automatically.

C. TASKER (WRITTEN): The author has given us some very interesting details of the heating of a modern Stratoliner for high altitude flying and the means taken to attain comfort for the occupants.

This paper would, however, be increased in value for those interested in the subject of *comfort* air conditioning if the author could give us details of the range of temperatures and humidities held in these ships. What has been found the most comfortable temperature by the majority of the passengers and how do the crew, who must be actively engaged mentally if not physically during the flight, react to the conditions which the comparatively inactive passengers prefer?

The hostess controls the distribution of the air and presumably, within limits, the temperature; but suppose half the passengers wish to have the maximum air recirculation effect and adjust the high velocity nozzles accordingly, what is the effect on the other passengers? Would this upset the control normally operated by the hostess?

Must the humidity be controlled for the sake of instruments, radio equipment, etc., and to avoid static, or is it allowed to vary as it may?

One interesting point on which the author could perhaps give very valuable information is as follows: Is there any difference between the conditions found most comfortable when operating up to 8,000 ft with no supercharging, at 8,000 to 15,000

ft when the pressure is held constant and at altitudes up to 20,000 ft when the differential pressure between the interior and the atmosphere is held at about 2½ lb per square inch. At these higher altitudes, when the air temperatures are low and therefore the absolute humidity low, what percentage humidity results in the cabin? Is artificial humidification necessary to increase comfort?

T. H. URDAHL (WRITTEN): It is of timely interest to the heating and ventilating engineer that in our latest mode of transportation such careful consideration is being given to the comfort of the operating personnel and the traveler. Comfort, as expressed in terms of physiological as well as psychological well-being, is second only to safety, and where safety is dependent upon human factors, it is therefore dependent upon comfort. Certainly, the attention of the best efforts of engineers, doctors, and physicists in cooperative action should be focussed upon the problems of the aircraft designer.

We can readily agree that the improvement of the airplane during the past 10 years has been phenomenal. However, in comparison with preceding centuries the gain in general knowledge concerning the factors applying to human comfort during the past 10 years may also truly seem phenomenal. Progress in all lines of endeavor has been directed toward the future rather than upon a continuation of the status quo based upon precedence which led to general and widespread economic collapse. Lessons most thoroughly taught by experience seem to be those gained at greatest cost of *trial and error,* but these are too frequently forgotten with the passing of time.

For progress in air travel those connected with this new industry have had uppermost in their minds the factor of safety, and comfort is a prime essential to the pilots of an airliner upon whose constant, alert ability so much depends. In a secondary measure comfort is vital to attracting the *pay load* which makes the operation of air lines possible.

Our railroads, by way of illustration, have demonstrated the fallacy of *trial and error* methods of producing and maintaining comfort conditions. After extensive expenditures and experimentation upon those whose patronage they wish to attract, they are only today attaining the desired results on certain lines. The air transport industry begins with a clean slate. It has few traditions to fight and has an appreciation of the costs and dangers involved in the *trial and error* method of the determination of vital factors. Trial in the research laboratory eliminates most of the error with safety and assurance at a fraction of the usually disproportionate cost where no research has established the new precedent. To this end we may look forward to a progressive solution of all the factors involved in comfort for high altitude flying through the cooperative effort of research personnel in the heating and ventilating and aircraft industries.

B. M. WOODS (WRITTEN): The material included in the discussion is well balanced. But the article contains too much detail on the mechanical operation of valves, blowers, etc., and not enough on the general methods employed.

Granting that the balance of material is proper, I feel that the detailed discussions of the control mechanisms lack clarity. In writing or talking about the minutiae of any mechanical device, it is extremely difficult to present a clear picture without the help of diagrams or sketches. In the present case, diagrams and sketches are scarce, and a normally careful study of the written words is insufficient to give a clear picture either of the mechanism itself or of the precise manner in which it functions. An example of the difficulties encountered is found in the statements in the paper that "It (the inlet valve) may also act as an inflow relief valve if the cabin pressure should tend to fall below the atmospheric pressure on the outside of the cabin"; also in "the outlet valve also acts as an inflow relief if the cabin pressure tends to fall below the atmospheric pressure on the outside of the cabin." One's immediate reaction to these two statements is that since one valve is designed under normal operation to admit air to the cabin, whereas the other is designed to allow air to exhaust from the cabin, it is difficult to realize how each can act as

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an inflow relief. It is possible that a careful study of these paragraphs together with the remaining material presented in the article, might clarify this seeming paradox, and yet it is doubtful whether the matter would be clear in the minds of those to whom such a paper is presented once verbally.

It is probable that the author will supplement the paper on presentation with comments or diagrams not reproduced in the article itself. In this case, those parts which seem hazy after a normal reading of the article will probably be clarified.

There are certain factors mentioned in the article which arouse interest in the mind of the reader but which are not discussed fully enough to answer the questions which they suggest. For example, it is stated that the cabin is designed to withstand 6 lb per square inch internal pressure. However, it further states that the maximum operating pressure used is two and a half pounds per square inch. Since this latter pressure provides an equivalent altitude of 11,000 ft when the airplane is flying at an actual altitude of 19,000 ft, the reader naturally wonders why higher internal pressures are not used in order to bring the equivalent altitude within the cabin down to a more comfortable level. It would be interesting to know why no attempt is made to lower the equivalent altitude since 11,000 ft is known to cause considerable discomfort to many air travelers.

It is stated in more than one place in the article that there is a rise in temperatures due to compression of the air being forced into the cabin. The reader would presume that this rise is considerable, since the use of an inter-cooler is mentioned. It would be interesting to know how much increase in temperature takes place due to compression, and, since there have been indications that the rise is considerable, to know what type of inter-cooler is used.

Fig. 7 is not clearly explained. Why is it that the obtainable pressure differential between the inside and outside of the cabin is apparently greater at 1200 rpm engine speed than at 1800 rpm engine speed? Perhaps I have misinterpreted the data presented on the graph, but the conditions just mentioned seem to exist, and, if they do, are possibly attributable to peculiar efficiency curves of centrifugal blowers. Some concrete statements concerning the facts presented in Fig. 7 would not only be interesting but desirable.

In spite of the questions raised, I believe the presentation timely and valuable. The Society is indebted to Mr. Tomlinson for an important contribution.

AUTHOR'S CLOSURE: I might reply somewhat to Mr. Tasker's comments. We endeavor to maintain the passenger cabin temperature at about 72 F. The pilot's cockpit generally runs about 10 deg less than that; sometimes it runs a whole lot less than that, because you want to remember that the nose of the airplane is right out in front, getting a blast of air at 200 mph, or better, and has been mentioned down as low as 40 below zero. It gets awfully cold up there in front, where you have glass windshield covering a considerable expanse of your cockpit. It is pretty hard to keep it warm under those circumstances.

We cannot do much about the humidity problem. The weight complexity of any humidifier in a plane is too much for us to tackle; we cannot stand the weight. The humidity in the Boeing will vary between that encountered at sea level down to a possible minimum of around 15 per cent at 20,000 ft.

I have flown in the Army's experimental pressure cabin plane up to 32,000 ft, and there were complete instruments to measure humidity temperature, and we noticed that there was no serious drop off in humidity until we passed about 20,000 ft. Then it began to drop off and at 30,000 ft we were down to an indicated humidity something like 3 per cent, and I was informed by the Army boys who had made long flights in the ship that they got pretty thirsty, and had to drink quite a bit of water to keep their throats at all moist, if they were up three or four hours on a job.

So, until we begin to operate any length of time above 20,000 ft, the humidity angle will not be a serious problem, and I might add it has no effect on instruments or radio whatever. I think that is about all I can offer.

No. 1171

FRICTION HEADS IN STANDARD SIX-INCH PIPE

By F. E. Giesecke* and J. S. Hopper,** College Station, Tex.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the Engineering Experiment Station of the Agricultural and Mechanical College of Texas.

Introduction

THE chart of friction heads in black iron pipe, published in the Heating, Ventilating, Air Conditioning Guide of the American Society of Heating and Ventilating Engineers, is based on previous experimental determinations.[1] In these earliest experiments determinations were limited to pipes ranging in size from ½-in. to 3-in. The friction heads per foot of pipe for these sizes were the following:

For—

½-in. pipe $h = 0.014335v^{1.776}$ ..... (1)
¾-in. pipe $h = 0.00855v^{1.812}$ ..... (2)
1 -in. pipe $h = 0.006371v^{1.7767}$ ..... (3)
1¼-in. pipe $h = 0.004675v^{1.756}$ ..... (4)
1½-in. pipe $h = 0.003228v^{1.7998}$ ..... (5)
2 -in. pipe $h = 0.00253v^{1.6787}$ ..... (6)
2½-in. pipe $h = 0.00253v^{1.735}$ ..... (7)
3 -in. pipe $h = 0.00166v^{1.679}$ ..... (8)

By plotting the coefficients and exponents of v in these eight equations on logarithmic paper, it was found that they could be combined into the following:

$$h = 0.00685 \frac{v^{1.793-0.04}}{d^{1.275}} \quad \text{..... (9)}$$

Where

h = the friction head in feet of 70 F water, when 70 F water is flowing in standard black pipe ranging in size from ½ in. to 3 in.
v = velocity of water in feet per second.
d = actual internal diameter of pipe in inches.

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** Assistant Professor of Mechanical Engineering, A. and M. College of Texas. Member of ASHVE.

[1] The Friction of Water in Pipes and Fittings, by F. E. Giesecke. (*Bulletin* No. 1759, University of Texas.)

Presented at the 47th Annual Meeting of the American Society of Heating and Ventilating Engineers, Kansas City, Mo., January, 1941.

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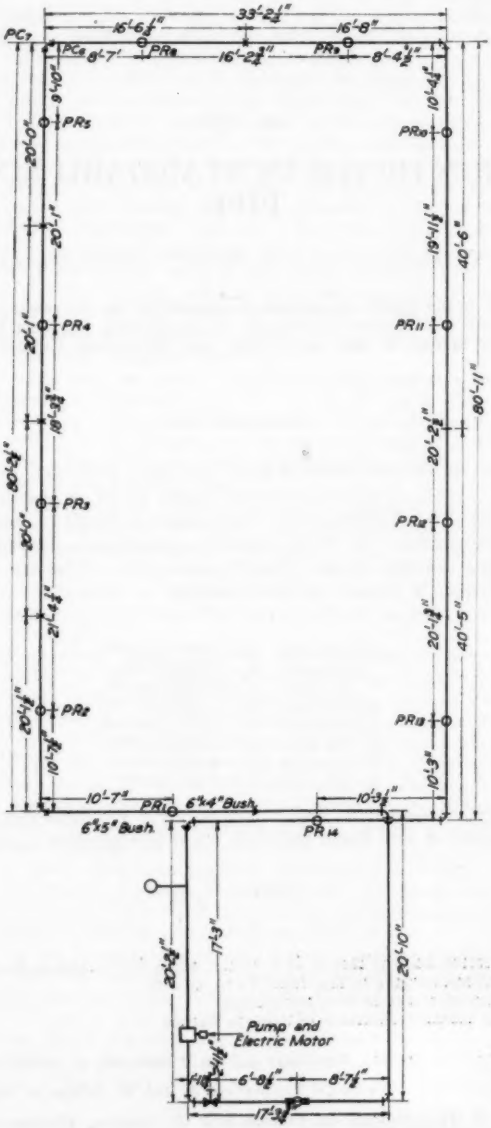

FIG. 1. PLAN OF 6-IN. PIPE SET-UP

In order to apply this equation to the design of hot water heating systems, tests were made to determine how the friction head is affected by the temperature of the water, and it was found [2] that it varies approximately as $t^{-0.19}$.

Equation (9) may therefore be changed to

$$h = \frac{0.01533 v^{1.79} d^{-0.04}}{t^{0.19}\, d^{1.275}} \qquad (10)$$

in which t represents the temperature of the water, in degrees Fahrenheit, flowing in the pipe and in which h is expressed in feet of water of the same temperature.

If the average temperature of the water in a hot water heating system is 140 F, the friction heads in the pipe lines of the system are

$$h = \frac{0.006 v^{1.79} d^{-0.04}}{d^{1.275}} \qquad (11)$$

so long as the pipes are new and clean.

In order to determine how the friction head in the pipe lines of a hot water heating system may be expected to increase during use, some pipes which had been in continuous use for about seven years were taken from the hot water heating system of the University greenhouse; and it was found that the coefficient of v in Equation (9) was about 17 per cent greater and the exponent about 6 per cent greater than in new and clean pipe.

Since the average temperature of water in a hot water heating system is generally somewhat higher than 140 F, the coefficient of v in Equation (11) was increased by 15 per cent (instead of 17 per cent) and the exponent by 5 per cent (instead of 6 per cent) producing the following:

$$h = 0.0069 \frac{v^{1.88} d^{-0.04}}{d^{1.275}} \qquad (12)$$

This equation was used to design the charts in THE GUIDE, 1940. The experimental data were determined for pipes ranging in size from ½ in. to 3 in. but the results seemed so consistent that it was considered safe to apply the resulting equation to pipes ranging in size up to 6 in.

When the chart was designed, hot water heating was not very popular and was applied, as a rule, only to small installations in which no large pipes were used. Since then, conditions have changed materially and many of the new heating installations are for hot water. Many installations are of considerable magnitude, requiring the use of large pipes, so it is important to know whether the charts in THE GUIDE, which are based on tests of small pipes, are sufficiently accurate to be used in the design of systems involving large pipes. Therefore, a Technical Advisory Committee [3] of the Society was organized and

[2] Loc. Cit. See Note 1.

[3] ASHVE Research Technical Advisory Committee on Flow of Fluids Through Pipes and Fittings: S. R. Lewis, *Chairman*; L. A. Cherry, G. C. Davis, T. M. Dugan, Earle W. Gray, R. T. Kern, H. A. Lockhart, Axel Marin, R. F. Taylor, E. L. Weber.

the program outlined provided for experimental determination of the loss of head in 6-in. standard iron pipe which is described in this paper.

OBJECTS

The principal object of the investigation was to determine the friction heads, for water of ordinary temperature, in:

1. New and clean standard 6-in. black pipe.
2. Screwed couplings of 6-in. pipe.
3. Welded joints of 6-in. pipe.
4. Screwed 90-deg elbows of 6-in. pipe.
5. Welded 90-deg elbows of 6-in. pipe.

A secondary object was to determine the feasibility of using welded elbows to meter the flow of liquids in pipes.

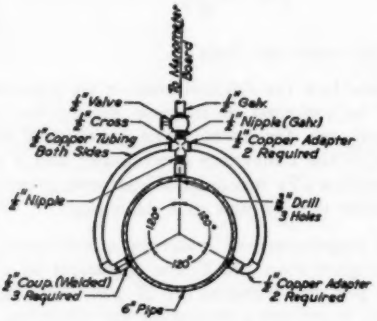

FIG. 2. PIEZOMETER RING

APPARATUS

The 6-in. pipe to be tested was supported on wooden blocks, near the floor of the laboratory of the Mechanical Engineering Department of the A. & M. College of Texas, as shown in Fig. 1. Fourteen piezometer stations were made in the pipe at the points marked PR1, PR2, etc., in Fig. 1. At each piezometer station were 3/16-in. holes, spaced 120 deg apart, around the pipe. The holes were drilled and the inner surface of the pipe adjacent to the holes was scraped with a special scraping tool so that no burr was left at any of the holes. The three holes were connected by means of a piezometer ring, as shown in Figs. 2 and 3. The piezometer rings were connected, by means of carefully graded ½-in. pipe lines, to glass piezometer tubes mounted on a board, as shown in Figs. 4 and 5. The piezometer stations were located so that the losses of head for the prevailing velocity in the several typical portions of the pipe line were indicated by the differences in pressure at the corresponding piezometer stations. These losses of head were shown by the differences in the elevations of the water in the corresponding piezometer tubes. For

example, some of the typical sections of the pipe line between piezometer stations are as follows:

PR1 to PR2 21 ft 2 in. of pipe and one welded elbow.
PR2 to PR3 21 ft 4¼ in. of pipe and one welded joint.
PR2 to PR5 59 ft 9 in. of pipe and two welded joints.
PR9 to PR10 18 ft 8¼ in. of pipe and one screwed elbow.
PR10 to PR11 19 ft 11½ in. of pipe.
PR10 to PR13 60 ft 3¾ in. of pipe and one screwed coupling.

WELDED JOINTS

In the welded pipe used in these tests two of the welds were position welds and were made in three passes with 5/32-in., 3/16-in., and 5/32-in. rods, used in the order named. All other welds were roll welds and were made with three beads as follows:

1. Stringer bead with a 5/32-in. rod.
2. Filler bead with a 3/16-in. rod.
3. Dresser bead with a 5/32-in. rod.

A 1/16-in. space was left between pipe ends. The pipe ends had a 45-deg bevel, which made the included angle rather large, but satisfactory results were secured with the metallic arc welding.

It is evident that all welds were good because the test failed to indicate a measurable loss of head at any one of the welded joints.

METHOD OF TESTING

The water was circulated through the pipe line by means of a 1300 gpm pump direct-connected to a 15 hp motor, as shown in Fig. 6. The variations in the volume of water with temperature changes were accommodated by an open expansion tank. The quantity of water circulated through the pipe line was measured by means of an orifice meter (Fig. 7) consisting of a monel metal diaphragm with a 2.76-in. orifice and constructed according to the directions.

The orifice meter was calibrated before the test was begun and again after the first seven series of tests had been completed. During the first calibration the water was pumped through the meter, in the same manner as during the 6-in. pipe test, and discharged into a weighing tank (Fig. 8). The time required to discharge 600 lb of water into a tank was determined by means of a stop-watch which was automatically started and stopped by the motion of the scale beam.

During the second calibration the water was allowed to flow through the meter from the College water mains and the time required to discharge 600 lb of water was determined by means of a stop-watch as before.

During both calibrations a buffer plate was suspended in the upper portion of the tank, so that the impact of the water on the tank and on the weighing

[4] See *Bulletin* No. 109, University of Illinois.

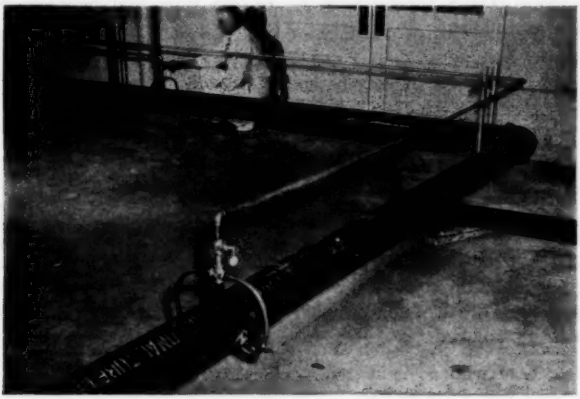

FIG. 3. PORTION OF 6-IN. PIPE WITH PIEZOMETER RING

platform was the same throughout the time during which water was being discharged into the tank.

The calibration curve derived in this manner is shown in Fig. 9. The points shown by circles were determined during the first calibration, and those shown by triangles, during the second calibration. The curve, which was drawn before the second calibration had been made, represented the points of the second calibration so well that no change was necessary after the second calibration.

In making a test run, the pump was started while the valve in the discharge line was closed (Fig. 6). The test was then begun either with the valve almost closed, so as to secure a low velocity (from 1 to 2 fps), or with the valve wide

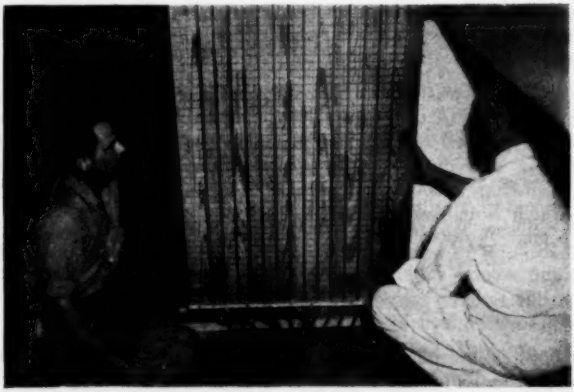

FIG. 4. PIEZOMETER TUBES

open, which gave a velocity of about 6 fps. The pipe system was purged of air by disconnection of the glass manometer tubes and allowing water to escape freely through the pipes leading to these tubes. After purging, the glass tubes were connected, and the water was allowed to circulate through the system until the water level in the manometer tubes became constant, at which time readings were taken at the flow meter and at the manometer tubes.

After the first set of readings had been taken, the valve setting was changed slightly to produce the next desired rate of flow. This process was repeated

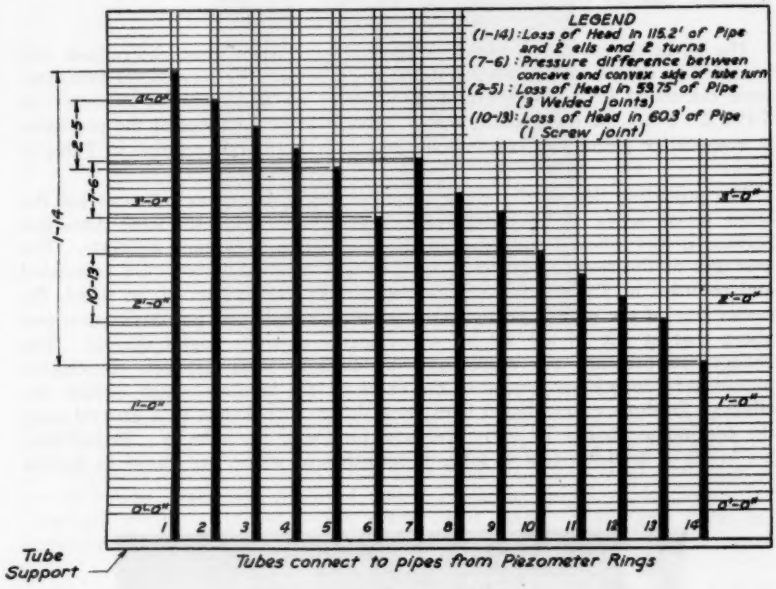

FIG. 5. PIEZOMETER TUBES

for different quantities of flow until the desired range of velocity had been attained in the 6-in. pipe. It required about three hours to complete such a run for each series of valve settings. Each series included about 14 sets of readings.

Nine series of tests were made. The data secured thereby were considered sufficient for the purpose of this investigation. The results of the fourth series are shown in Table 1, which is submitted here only as an illustration of the general method employed in recording and analyzing the data.

All calculations were made with the slide rule since it can be read as accurately as the orifice meter (Fig. 7) or the piezometer pressures (Fig. 4).

The values shown in Table 1 were transferred to Fig. 10 and lines were drawn through the resulting points to represent, as accurately as possible, the

averages of the observed values. The lines of Fig. 10 show that the flow of the water in the 6-in. pipe was turbulent for velocities higher than 3 fps, because, for these higher velocities, the friction heads are represented by straight lines when plotted to logarithmic coordinates against the corresponding velocities.

It is interesting to note that the results of this series of tests, as shown in Fig. 10, indicate that the transition from laminar to turbulent flow ended at a velocity of about 3 fps. The temperature of the water was about 90 F and Reynolds' number was approximately 194,000.

The equations of the straight portions of the lines were determined and recorded as shown in Fig. 10. With these equations, the piezometer pressures were calculated for velocities of 3, 4, 5, and 6 fps and recorded as shown in Table 2. From these values, shown in Table 2, the differences of the pressures at consecutive piezometer connections were calculated and recorded in Table 3.

The procedure described for the fourth series of tests was followed for the other seven series. The results of the first series of tests were discarded because it was believed that those of the later series were more accurate. The averages of the results secured for the remaining eight series were calculated and recorded in Table 4 and represented graphically in Fig. 11, in which the center line of the pipe is taken as the coordinate axis and the static pressures above that at PR 14 are set off as ordinates at right angles thereto. The changes in pressure are represented by straight lines between consecutive piezometer connections except in the cases of the sections which include the elbows. In those cases, dotted lines were drawn to indicate, in a general way, the continuous change in static pressure, for the 6 fps velocity. In addition, the losses of head per foot of pipe, in milinches of water, are shown by figures

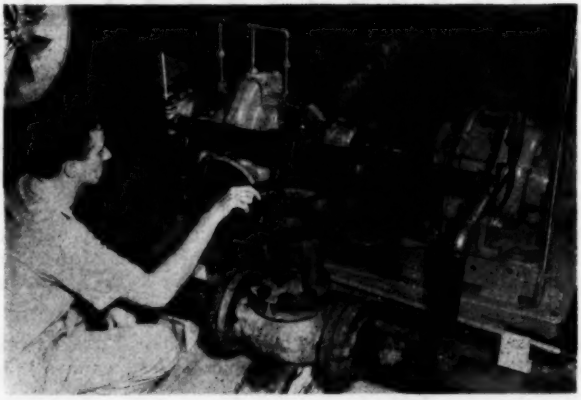

FIG. 6. MOTOR, PUMP, AND VALVE

FIG. 7. ORIFICE METER

on the respective lines for each of the seven sections and for velocities of 3, 4, 5, and 6 fps. It appears from Table 4 and from Fig. 11 that the total friction head for the entire pipe line, consisting of 215.18 ft of 6-in. pipe, 2 welded elbows, 2 screwed elbows, 1 screwed coupling, and 4 welded joints, for velocities of 3, 4, 5, and 6 fps, were 14.70, 25.07, 37.86, and 52.66 in., respectively.

Having these data, the problem is to determine the friction heads in 1 ft of pipe, 1 screwed elbow, 1 welded elbow, 1 screwed coupling, and 1 welded joint, for the several velocities.

When this study was planned it was assumed:

1. That the piezometer connections were far enough from the adjacent elbows that the influence of the turbulence within the elbow would not extend beyond the nearest piezometer connection.

2. That the unit losses of head in Sections 10-11 and 12-13 would be the lowest of any in the pipe line and could be used as the friction head in one foot of pipe, because those two sections contain no pipe fittings or joints.

3. That the unit loss of head in Section 11-12 would be slightly higher than those in the adjoining sections and that the difference could be used to calculate the friction head of one screwed coupling.

4. That the unit losses of head in Sections 2-3, 3-4, and 4-5 would be slightly higher than those in Sections 10-11 and 12-13 and that the differences could be used to calculate the friction head of one welded joint.

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TABLE 1—OBSERVED DATA IN SERIES 4[a]

ORIFICE METER IN HG	PIPE WATER TEMP. DEG F	WATER CU FT PER SEC	TUBE WATER TEMP. DEG F	STATIC PRESSURE READINGS—INCHES OF WATER—FOR PIEZOMETER TUBES (*Numbered as Shown*)														WATER VELOCITY FT-SEC
				1	2	3	4	5	6	7	8	9	10	11	12	13	14	
26.4	87.2	1.24	82.0	77.5	72.0	67.5	63.5	59.7	50.0	61.2	54.3	50.7	43.8	39.1	34.9	30.5	22.7	6.17
24.4	87.7	1.18	82.0	72.3	66.8	62.8	58.5	55.2	46.0	56.2	50.6	47.5	40.5	36.3	32.5	28.4	21.2	5.88
22.3	88.2	1.12	82.0	66.4	61.2	58.0	54.5	51.5	43.0	52.3	47.0	43.9	37.8	33.8	30.1	26.4	19.5	5.58
19.7	88.8	1.05	82.0	59.5	55.0	52.0	49.0	45.5	38.2	47.0	42.0	39.1	33.8	30.3	27.1	23.8	17.9	5.23
17.9	89.2	1.00	82.5	54.3	50.8	47.5	44.5	42.2	35.5	43.0	38.5	36.0	29.1	26.0	25.0	21.9	16.7	4.98
16.3	89.5	0.95	82.5	50.8	46.8	44.0	41.2	39.0	32.8	39.8	35.7	33.5	29.0	25.9	23.2	20.5	15.7	4.73
14.2	89.8	0.89	83.0	44.8	41.5	39.2	36.8	34.8	29.0	35.4	31.7	29.8	26.2	23.3	20.9	18.6	14.1	4.43
12.3	90.1	0.83	83.0	39.8	36.7	34.8	32.9	30.7	25.8	31.2	28.2	26.5	23.3	20.8	18.8	16.7	12.9	4.14
10.5	90.3	0.77	83.0	34.6	32.2	30.0	28.8	27.1	23.0	27.7	24.8	23.2	20.8	18.6	16.9	14.9	11.7	3.84
8.8	91.0	0.71	83.0	29.8	28.0	26 3	24.6	23.2	20.0	24.0	21.7	20.3	18.2	16.2	14.8	13.2	10.4	3.54
7.0	91.3	0.64	83.5	25.2	23.8	22.2	21.0	19.7	17.0	20.3	18.4	17.5	15.8	14.0	12.9	11.6	9.4	3.19
5.6	91.7	0.57	83.5	20.8	19.5	18.6	17.8	16.8	14.6	17.0	15.7	14.8	13.6	12.1	11.2	10.1	8.3	2.84
3.8	92.1	0.47	83.5	16.0	15.0	14.5	13.3	12.7	11.7	13.0	12.1	11.7	10.8	9.7	9.1	8.2	7.0	2.34
2.4	92.6	0.37	83.5	12.0	11.3	10.9	10.2	10.0	9.0	10.1	9.3	9.0	8.8	7.8	7.4	6.9	6.1	1.84
Total				603.8	560.6	528.3	496.6	468.1	395.6	478.2	420.7	403.5	351.5	313.9	284.8	251.7	193.6	
Average				43.1	40.0	37.7	35.5	33.45	28.2	34.2	30.05	28.8	25.1	22.4	20.35	17.95	13.8	

[a] Tables like this one, which shows the experimental results of Series 4, were prepared for the other nine series but are not reproduced in this paper.

5. That the unit losses of head in Sections 1-2 and 5-8, would be higher than those in Sections 10-11 and 12-13, and that the difference could be used to calculate the friction head of one welded elbow.

6. That the unit losses of head in Sections 9-10 and 13-14 would be higher than those in Sections 10-11 and 12-13 and that the difference could be used to calculate the friction head of one screwed elbow.

The results of the experiment, recorded in Fig. 11, show that these assumptions were erroneous. The unit losses of head in Sections 10-11 and 12-13 are practically equal to each other but they are higher, instead of lower, than

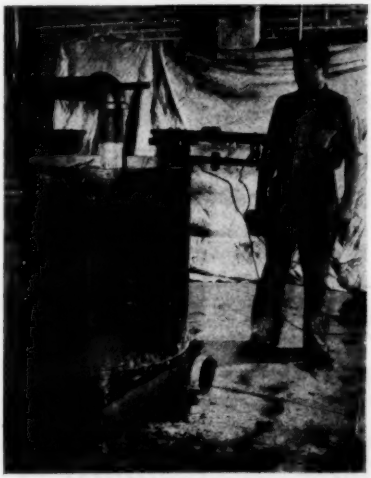

FIG. 8. WEIGHING TANK IN LABORATORY

the unit loss of head in Section 11-12. The only apparent explanation of this condition is that the turbulence in Elbow 3 extended beyond PR 10 and that the turbulence in Elbow 4 extended beyond PR 13. It seems reasonable that the influence of Elbow 3 should have extended into its downstream tangent beyond PR 10, but it does not seem reasonable that the influence of Elbow 4 should have extended into its upstream tangent beyond PR 13. However, no other explanation is apparent at this time.

The result of this situation is that the unit losses of head in Sections 10-11 and 12-13 cannot be considered the unit friction heads in 6-in. pipe.

In Sections 2-3, 3-4, and 4-5 the unit losses of head are very nearly alike; each section contains one welded joint and slight differences in the welded joints may account for the slight differences in unit head losses. It seems safe to conclude that the influence of the turbulences in Elbows 1 and 2 did not extend beyond the nearest piezometer connections. This seems reasonable because Elbows 1 and 2 are welded and have the same internal diameter as the

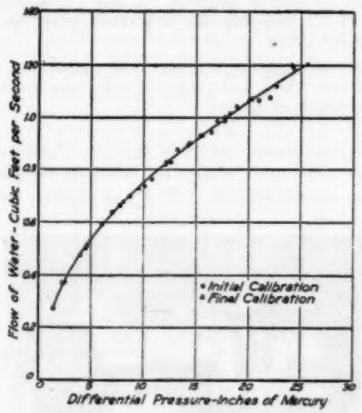

FIG. 9. CALIBRATION CURVE

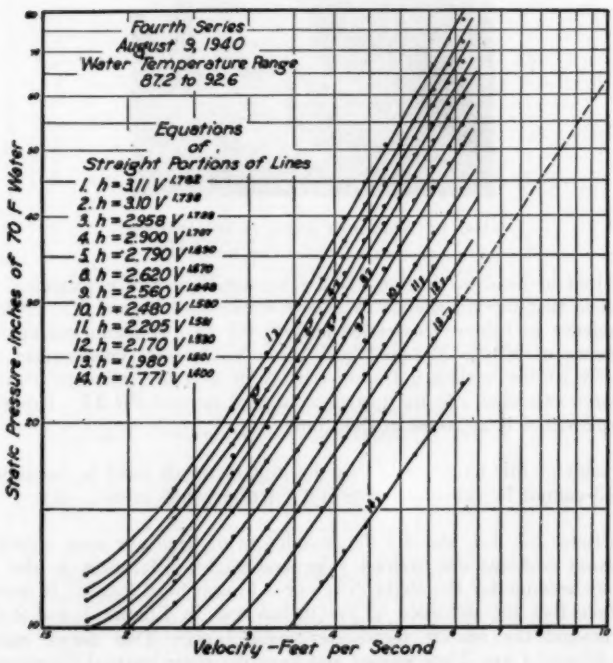
Fourth Series
August 9, 1940
Water Temperature Range
87.2 to 92.6
Equations of Straight Portions of Lines
1. h = 3.11 V^1.782
2. h = 3.10 V^1.798
3. h = 2.958 V^1.788
4. h = 2.900 V^1.787
5. h = 2.790 V^1.830
6. h = 2.620 V^1.870
9. h = 2.560 V^1.868
10. h = 2.480 V^1.880
11. h = 2.205 V^1.881
12. h = 2.170 V^1.930
13. h = 1.980 V^1.881
14. h = 1.771 V^1.920
Static Pressure - Inches of 70 F Water
Velocity - Feet per Second

FIG. 10. STATIC PRESSURES OF SERIES 4

TABLE 2—STATIC PRESSURES—SERIES 4
Inches of 70 F Water

PIEZOMETER STATIONS	WATER VELOCITY—FPS			
	3	4	5	6
1	22.00	36.80	54.60	75.80
2	20.82	34.35	50.75	69.60
3	17.82	28.95	42.20	57.40
4	16.41	26.55	38.45	52.10
5	15.60	25.05	36.25	48.80
6	14.08	22.10	31.53	42.10
7	12.52	19.75	28.05	37.40
8	16.40	26.60	38.50	52.10
9	15.61	25.10	36.25	48.98
10	11.63	18.02	25.45	33.60
11	12.55	19.75	28.10	37.45
12	11.64	18.10	25.48	33.60
13	10.28	15.82	22.16	29.20
14	8.25	12.33	16.82	21.77

pipe; whereas Elbows 3 and 4 are screwed elbows and have a larger internal diameter than the pipe and therefore create more turbulence than the welded elbows.

It seems permissible, therefore, to consider the unit losses of head in Sections 2-3, 3-4, and 4-5, as the friction head in 1 ft of a 6-in. pipe line with a normal number of good welded joints. If the weighted mean values of these unit head losses are plotted on logarithmic paper, it will be found that the unit friction-head is

$$h = 0.00725\ v^{1.848} \quad \ldots\ldots (13)$$

where v is in feet per second and where h represents the friction head in inches of 70 F water, per foot of pipe, when water at 95 F is flowing in the pipe.

The friction heads in the two welded elbows can then be calculated by adding the losses of head in Section 1-2 and those in Section 5-8 for velocities of 3, 4, 5, and 6 fps, and subtracting from the sum the losses of head calculated by Equation (13) for the straight pipe in those two sections, 39.63 ft, for the same velocities and plotting the results on logarithmic paper. This will show that the friction head of one welded 6-in. elbow is

$$h = 0.023\ v^{2.3} \quad \ldots\ldots (14)$$

Instead of using only the losses of head found in Sections 2-3, 3-4, and 4-5 to determine the friction head in 1 ft of a welded pipe line, as described previously, it may be better to use the losses of head in the entire straight portion of the experimental pipe line, *i.e.*, Sections 2-5, 8-9, and 10-13, as representing more nearly the type of pipe line ordinarily installed in practice.

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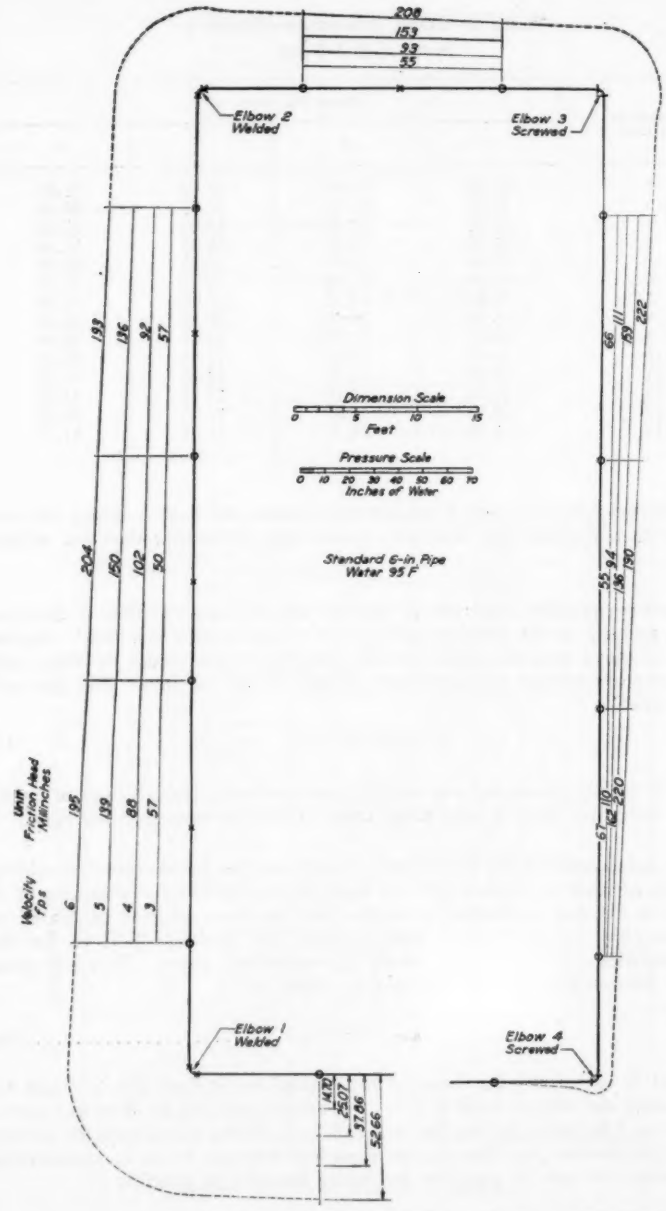

FIG. 11. LOSS OF HEAD IN 6-IN. PIPE AND FITTINGS

TABLE 3—STATIC PRESSURE DIFFERENCES—SERIES 4
Inches of 70 F *Water*

PIEZOMETER STATIONS	WATER VELOCITY—FPS			
	3	4	5	6
1–14	13.75	24.47	37.78	54.03
1–2	1.18	2.45	3.85	6.20
2–3	1.09	1.94	3.20	4.70
3–4	0.81	1.60	2.55	3.30
4–5	1.02	1.81	2.85	4.26
5–8	1.41	2.40	3.75	5.30
8–9	0.81	1.50	2.20	3.30
9–10	1.52	2.95	4.72	6.70
10–11	1.55	2.45	3.42	4.65
11–12	0.89	1.73	2.60	3.80
12–13	1.35	2.20	3.29	4.40
13–14	2.03	3.49	5.34	7.43

If this is done, the unit friction heads will be

$$h = 0.00834\, v^{1.78} \quad \ldots\ldots (15)$$

for one foot of 6-in. pipe;

$$h = 0.02127\, v^{2.275} \quad \ldots\ldots (16)$$

for one welded 6-in. elbow; and

$$h = 0.1101\, v^{1.873} \quad \ldots\ldots (17)$$

for one screwed 6-in. elbow, as shown in Fig. 12, where v is in feet per second and h represents the friction head, in inches of 70 F water when water at 95 F is flowing in the pipe and fittings.

In the calculation to determine the friction heads in screwed or welded elbows, it should be noted that the length of pipe connected to the elbow was

TABLE 4—STATIC PRESSURE DIFFERENCES—SERIES 2,3,4,5,6,7,8,9
Inches of 70 F *Water*

PIEZOMETER STATIONS	VELOCITY—FPS				SECTION OF PIPE LINE
	3	4	5	6	
1–14	14.70	25.07	37.86	52.66	215.18 ft pipe + 2 welded elbows + 2 elbows + 4 welds + 1 coupling
1–2	1.52	2.62	3.98	5.68	21.17 ft pipe + 1 welded elbow
2–3	1.22	1.88	2.96	4.16	21.38 ft pipe + 1 weld
3–4	0.92	1.86	2.74	3.74	18.31 ft pipe + 1 weld
4–5	1.15	1.85	2.73	3.87	20.08 ft pipe + 1 weld
5–8	1.24	2.22	3.44	4.89	18.42 ft pipe + 1 welded elbow
8–9	0.89	1.51	2.49	3.38	16.23 ft pipe + 1 weld
9–10	1.87	3.21	4.97	6.92	18.73 ft pipe + 1 elbow
10–11	1.31	2.21	3.18	4.42	19.96 ft pipe
11–12	1.12	1.89	2.78	3.85	20.21 ft pipe + 1 coupling
12–13	1.35	2.22	3.27	4.42	20.15 ft pipe
13–14	2.11	3.60	5.32	7.33	20.54 ft pipe + 1 elbow

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measured from the piezometer station to the point of intersection of the center lines of the pipe. Consequently, the reported friction head in one elbow is less than it would be if only the actual length of pipe, from the piezometer station to the elbow, had been used in the calculations. The foregoing method was adopted because it is believed to be common practice to measure pipe lines between the intersections of center lines of pipes, and because it simplifies friction head calculations materially. This method reduces an elbow to a *no-length* fitting.[5]

To compare the results of Equation (15) with those of Equation (9) and with those shown by the chart in THE GUIDE, Equation (15) should be modified so as to apply to 70 F water instead of to 95 F water. Assuming that, for this particular range of temperatures, the friction head varies inversely as $F^{0.19}$, the modified form of Equation (15) will be

$$h = 0.0089\, v^{1.785} \quad \ldots\ldots (18)$$

The modified form of Equation (9) to apply to 6-in. pipe and to use inches instead of feet as a unit of measurement, is

$$h = 0.00826\, v^{1.885} \quad \ldots\ldots (19)$$

As an additional check on the experimental results, the losses of head were calculated according to the formula given by Nikuradse[6] for smooth pipe. This formula, with slight transformations, expressed in English units, and for values of R above 100,000 is

$$h = \left(0.0384 + \frac{2.652}{R^{0.237}}\right) \frac{L}{d} \frac{v^2}{2g} \text{ in.} \quad \ldots\ldots (20)$$

The friction heads, according to the three equations, and according to the chart in THE GUIDE 1940 are:

	VELOCITIES			
	3	4	5	6
Equation 19 (9)	51	83	121	163
GUIDE, Equation 12	58	96	141	196
Equation 18 (15)	63	105	157	218
Nikuradse	46	77	115	160

According to Equation (20), the loss of head in 6-in. pipe, for velocities ranging from 3 to 6 fps, is about 12 per cent lower for 140 F water and about 20 per cent lower for 210 F water than for 70 F water.

In comparing the above values, it should be borne in mind that the values calculated by means of Equations (18) and (19) are for 70 F water, while those from THE GUIDE are for 140 F water and include an allowance for incrustation.

It is evident from this study that the charts in THE GUIDE should be revised,

[5] The Friction of Water in Elbows, by F. E. Giesecke, C. P. Reming, and J. W. Knudson, Jr. (*Bulletin* No. 2712, University of Texas).
[6] *Forschungsheft*, by Nikuradse, 356, 1932.

as soon as definite information regarding the influence of temperature and of incrustation from corrosion on the friction heads is available.

WELDED ELBOW AS A METER

The possibility of using a welded elbow to measure the flow of liquids in pipes was checked. The differences between the static pressures at piezometers

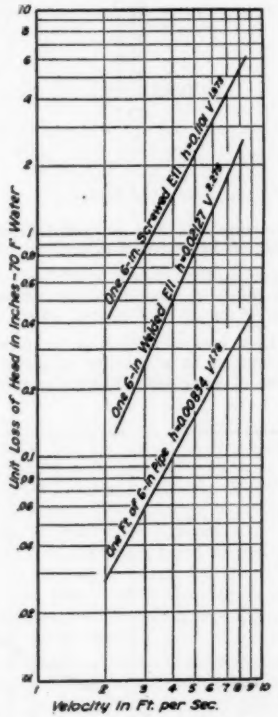
Unit Loss of Head in Inches - 70° F. Water
Velocity in Ft. per Sec.
One 6-in. Screwed Ell. h=0.1104 V 1.978
One 6-in. Welded Ell. h=0.0247 V 2.378
One Ft. of 6-in. Pipe h=0.00834 V 1.78

FIG. 12. UNIT LOSS OF HEAD IN 6-IN. PIPE AND ELBOWS

6 and 7 were plotted in Fig. 13 against the volume of water flowing in the pipe, in cubic feet per second. The volume of water was determined from the readings of the orifice meter.

It appears from the graph that a welded elbow can be used as a meter; it also appears that the difference between the pressures on opposite sides of the elbow is an exponential function of the velocity, since the pressure difference plotted against volume is a straight line on logarithmic paper.

TABLE 5—WATER TEMPERATURES FAHRENHEIT AT BEGINNINGS AND ENDS OF SERIES

SERIES	BEGINNING	END
2	102.0	107.5
3	90.0	96.0
4	87.2	92.6
5	86.0	95.6
6	87.1	92.7
7	93.2	98.0
8	93.6	97.8
9	97.7	100.9
Average.....	92.1	97.6

CONCLUSIONS

1. The friction heads in couplings and in well-fabricated welds are very small and may be neglected in friction head calculation.

2. The friction head is one foot of a new and clean standard black 6-in. pipeline with a normal number of joints or couplings for water having a temperature of about 95 F is $h = 0.00834\, v^{1.78}$ in. of 70 F water, where v is in feet per second.

3. The transition period from laminated flow to turbulent flow in the 6-in. pipe line shown in Fig. 1 seems to extend to a velocity of about 3 fps. Additional studies should be made of the losses of head for velocities in this transition range.

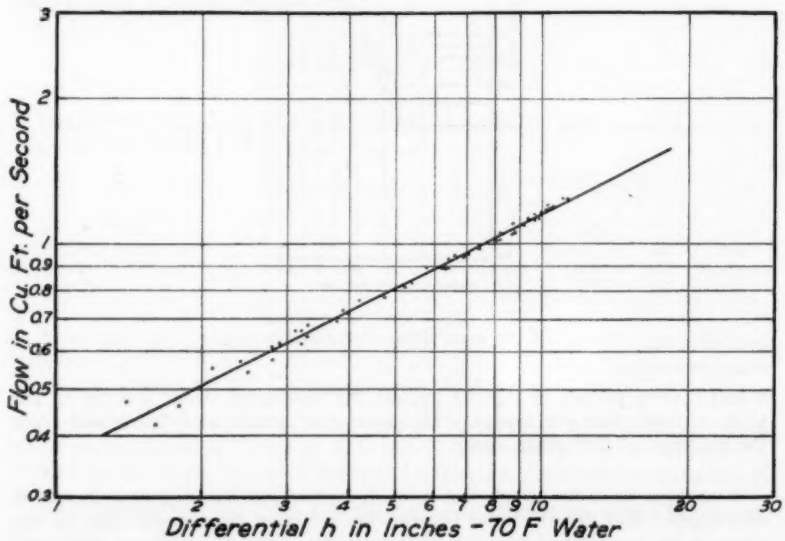

FIG. 13. RELATION OF FLOW AND PRESSURE DIFFERENCE AT PIEZOMETERS 6 AND 7

4. The friction head in one 6-in. screwed elbow is $h = 0.1101\, v^{1.873}$ in. of 70 F water, where v is in feet per second, when the temperature of the water flowing through the elbow is about 95 F and when the lengths of the connecting pipe lines are measured to the intersection of their axes. For the same conditions, the friction head in one welded elbow is $h = 0.02127\, v^{2.275}$ in. of 70 F water.

5. A welded elbow can be used as a flow meter by measuring the difference in the static pressures between the concave and the convex sides and at the center of the elbow. This measurement is preferably made by means of an inclined manometer.

6. Additional research should be conducted to determine the effect of water temperature and also the effect of incrustation resulting from corrosion on the friction head in pipes and fittings.

ACKNOWLEDGMENTS

This investigation was sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and conducted at the Engineering Experiment Station, Texas A. and M. College, College Station, and acknowledgment is given for the assistance of the following: Bell and Gossett Co., A. M. Byers Co., Chicago Pump Co., Crane Co., Grinnell Co., Hoffman Specialty Co., Kewanee Boiler Corp., Minneapolis-Honeywell Regulator Co., National Tube Co., Nash Engineering Co., and Weil McLain Co. Acknowledgment is also given to Robert Cook, instructor in welding at A. & M. College of Texas, who did all the welding.

DISCUSSION

H. A. LOCKHART (WRITTEN): Hot water heating has changed its character in the past few years due to the advent of equipment built primarily for hot water heating, and there has been a vast expansion in viewpoint, allowing the use of hot water heating on almost any size project. As a result, the Industry has had to work with information which has been inadequate because the technological advance has not been as rapid as the expanded use of hot water heating.

The relationship shown between the information as determined by this test and previous information in THE GUIDE indicates that the charts which have been published previously in THE GUIDE have been well within the range of good practice. It is impossible, even with the additional information, to calculate a system so closely that the experimental results will agree entirely with the calculated results. It will be necessary for us to do in the future as we have done in the past, and that is, to allow our engineering factor of safety so that we can be assured of proper results.

It would seem to me that the results of this paper indicate that further work of a similar nature should be undertaken in bringing the engineering data up to date.

The results so far on welded and screwed elbows, indicate to us that this variation will be true in the other sizes of pipes that are used. From experimental data which we have, there is an indication that other equipment installed in a hot water heating system has been changed sufficiently to warrant questioning of the present values. We would like to consider this investigation as the beginning of further research work to bring right up to date the information necessary to properly design a hot-water heating system.

FERDINAND JEHLE (WRITTEN): This paper has been needed for a long time to set our minds at rest on the question *Can the performance of the smaller pipe sizes predict what will happen in the larger sizes?* Like so many pieces of research work, this report has brought out our lack of exact similar knowledge of other devices. I refer to devices which must be used with pipes in order to complete a hot water

heating system. Particularly, do I refer to so-called flow fittings of various manufacturers. Manufacturers of these devices do not give in a standardized way, exact information as to the pressure drop across such fittings, but it is hinted that it might be many times that existing across several feet of pipe. The need for a simple test made with standardized equipment is, therefore, apparent and I believe that the Society should, through its Committee on Research, prescribe such a test procedure to be followed by manufacturers of any and all devices which affect the resistance to water flow in heating systems.

In addition to the resistance characteristics of the circuit, more exact knowledge of the pumps (circulators) would not be amiss. The Society might, to advantage, also adopt a simple testing code for circulating pumps which are now in such general use.

C. R. COX[†] (WRITTEN): It is self-evident that the authors planned the test program with utmost logic and executed it in a manner beyond reproach. Their interpretation of the results is clearly stated and should prove of great value to those engineers who are confronted with the problem of fluid flow through standard pipe.

J. H. VAN ALSBURG: I would like to know how the results based on this 6-in. pipe compare with the results now appearing in THE GUIDE?

DR. GIESECKE: The experimental results for the 6-in. pipe are slightly higher than those shown in THE GUIDE. However, as shown in Fig. 11, there was a considerable variation along the 215-ft pipe line, in the unit loss of head. This variation was probably the result of variations in the welded joints and of variations in the effect of the elbows on the loss of head in the adjacent pipe. The loss of head, per foot of pipe, shown in Fig. 12, is the average unit loss of head for the entire pipe line. I believe the values shown in THE GUIDE should be increased slightly for the larger pipes.

AXEL MARIN: On the piezometric diagram, Fig. 2, I notice 3/16-in. holes used connecting the ring to the pipe. I am wondering if those are not a bit large. Can you be certain you do not actually have circulation around through that ring. The readings obtained were somewhat at variance with what you expected. Were you not probably getting some dynamic head instead of merely static?

Another point, Fig. 10 shows static pressures plotted against velocity in feet per second. I notice in the results, that Reynolds' Number for about 3 fps is 194,000. At the critical point the value is 2300, as I recall it, for a velocity in a 6-in. pipe corresponding to laminar flow, which is less than 1.25 centimeters per second.

I am wondering if the readings from the Piezometer rings might not be taken with smaller holes? Some years ago on fan tests, which were not far different from this, there was always a question why the flow did not check on the suction side as well as the other side. The two would check exactly if straighteners were used on the suction side.

Every installation has its own peculiarities. Why not test a long length of pipe with no joints or use welded joints, which simulates a continuous pipe, because if the walls are well done the surfaces are smooth; then take elbows in another set-up, subtract the loss of straight pipe from the pipe with elbows to ascertain the elbow loss.

DR. GIESECKE: The size of the openings is 3/16 in.; there are three openings, spaced 120 deg apart, at every piezometer connection. We believe this arrangement gives satisfactory results.

The transition stage, from laminar to turbulent flow, extended to much higher velocities than expected. During laminar flow, the loss of head varies directly as the velocity; this variation is shown, with logarithmic coordinates, by a straight line making an angle of 45 deg with the horizontal. During steady turbulent flow, the

[†] Vice-President, National Tube Co., Pittsburgh, Pa.

loss of head varies as a power of the velocity higher than one and lower than two; for a 6-in. pipe, for example, it varies as the 1.78 power of the velocity; this variation is shown with logarithmic coordinates, by a straight line making an angle with the horizontal, whose tangent is 1.78, as shown in Fig. 10. During the transition period, when the flow has ceased to be laminar and is changing gradually to the steady turbulent state, there is a gradual change in the unit loss of head; this varying relation of loss of head to velocity is shown, with logarithmic coordinates, by a curved line tangent to the two straight lines mentioned above. The upper one of the two points of tangency occurred at a much higher velocity than was expected. The lower point of tangency was not reached in this investigation because the study did not extend into the range of laminar flow.

PROFESSOR MARIN: May I suggest then, on this section, that if it is possible, use only one connection, to see what the results will be, which should answer our question. This section, because of the turbulence may give flow through the ring. With laminar flow the static conditions are the same and there is no flow. This connection probably has a flow toward that opening. I have seen many cases where there is actually a flow right around the ring.

DR. GIESECKE: With air or water?

PROFESSOR MARIN: Both.

DR. GIESECKE: With water?

PROFESSOR MARIN: Yes, in the case of three connections.

DR. GIESECKE: I can think of nothing that might cause the flow of water from the 6-in. pipe into the ½-in. curved tube and then back into the 6-in. pipe except burrs on the sides of the 3/16-in. openings of such sizes and in such locations as to deflect the flow of water from the 6-in. pipe into the ½-in. tube. Professor Hopper was very careful to see that no burrs existed at any of the openings in the 6-in. pipe.

S. H. DOWNS: I think the point Dr. Giesecke brings out, about the curves in Fig. 10 curving, is a very important point. I do not believe he has investigated it far enough to satisfy the people who read this paper. You hear a lot of references to Reynolds' Number, but hear very few references where anybody has gone back and read his original paper and actually investigated what he said.

I would suggest, then, that this work be gone into further along that line, after getting Professor Reynolds' original paper. I think you will find he never said at any time that transition from laminar flow to turbulent flow is a point. He very definitely proved (the first one to do it) there was a range; that the range varied depending on which direction you were changing the velocity. If you change the velocity from a low value up to a high value, you obtain one range of transition. If you change the velocity in the reverse direction, from high to low value, you get a different range.

In this case the first thing you have to investigate is which way you were changing the velocity, and then, second, I think you should very definitely check against the actual values Professor Reynolds obtained himself, and not some of the later papers which I think quite often have misquoted him.

DR. GIESECKE: Mr. Downs is correct. The velocity at which laminar flow changes to turbulent flow is higher than the velocity at which turbulent flow changes to laminar flow.

The velocities at which these changes take place are quite well known for small pipe, but the results secured during this 6-in. pipe test indicate that further studies should be made to determine the corresponding velocities for the larger pipes.

This information would be of value in the design of hydraulic installations in which velocities lower than 3 fps are used in the larger pipes.

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R. E. MOORE: This paper is extremely interesting and certainly some important points have been discussed. We have at various times, talked about the system of forced hot water and the distribution of water as a heating medium, but I believe that we have overlooked a very important point and that is, *where* the heat comes from, the boiler itself.

In looking over most boilers today, you will find that the supply openings are out of the top, in the center of the boiler, or there are other openings located at random without thought of where the return from the heating system will come in. I have observed on many boilers, that placing a thermometer on the back section and on the front section will show a wide variance of temperature, denoting that internal circulation is not even and the greatest efficiency is not being obtained.

It is my opinion, and I believe I am expressing the thoughts of those engaged in the business of promoting the sale of forced hot water as a heating medium, that some fundamental research should be done on the circulation of water through the boiler itself. To my knowledge, there has never been a test run on a boiler to see what it will produce in efficiency when water is being pumped through it. Virtually all tests on standard cast-iron and steel-heating boilers have been made on how much steam they will produce, the condensation then being weighed and an empirical factor being used in determining the boiler's capacity when used as a part of a hot-water heating system.

I believe an excellent job has been done in regard to pumps, flow control valves, fittings and controls, but we have overlooked getting the ultimate from the fundamental source of heat, the boiler itself. Research work on boilers along this line will be a great help to our industry.

No. 1172

RADIATION AS A FACTOR IN THE SENSATION OF WARMTH

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

ONE of the most important fundamentals necessary for the application of comfort air conditioning is knowledge concerning the relation between the feeling of warmth of a person and his atmospheric environment. The atmospheric conditions affecting the feeling of warmth are the temperature of the air, its moisture content, and its movement. The relative importance of these factors is shown in the effective temperature scale, developed as one of the most important research investigations of the ASHVE Research Laboratory.

The fourth environmental factor influencing a person's feeling of warmth is radiation. The study of its significance was not included in the earlier research for the reason that it did not seem to be important in the American heating practice at that time, and it was, therefore, controlled by so insulating the rooms in which the studies were conducted, that the wall surfaces in view of the occupants were at approximately the same temperature as the air. With the growing adaptation of radiation to the American air conditioning practice, a technical advisory committee † was appointed during 1938 for the purpose of investigating some phases of this subject with the aid of a special grant of funds from the John B. Pierce Foundation. A comprehensive plan for carrying out the study was developed and a building consisting of two 12-ft x 15-ft x 8½-ft rooms connected by an observation room was built during 1939, in which an important phase of the investigations was carried out during the winter of 1939-40.

The results of the study to date indicate the degree to which the dry-bulb

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Presented at the 47th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., January, 1941.

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temperature, the moisture content, or the effective temperature of the environmental air must be decreased in order to compensate for the increased feeling of warmth resulting from a simultaneous increase in the radiation intensity, or vice versa.

TEST ARRANGEMENTS

A photograph of the test house located on the roof of the Bureau of Mines Warehouse Building is shown in Fig. 1. A floor plan of the test and observation rooms showing a number of the salient arrangements is given in Fig. 2. The shape and orientation of the building was designed to give as nearly as possible the same exposure to the walls of both rooms. As indicated in Fig. 2.

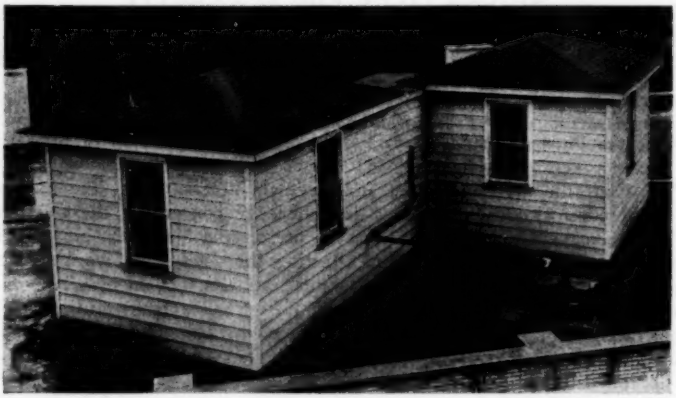

FIG. 1. VIEW OF RADIATION AND COMFORT TEST BUILDING

the two rooms are identified as the East and West rooms. Characteristics of the construction in different parts of the test house are given in Table 1.

In order to make the surroundings of the study as nearly typical of American heating practice as possible, the two rooms were so equipped that either might be heated by typical commercial hot water radiators or by a warm air circulation system. In Fig. 2 only one system of heating is shown in each room although during the testing program changes in the manner of heating for each room, that is, by radiation or convection, were made. In a given test the rooms are further identified as the radiation and convection rooms, sometimes abbreviated as *R* and *C*. Thus, in the arrangement in Fig. 2 is the west radiation room and the east convection room.

The heating requirements of both rooms were provided electrically so as to make available information concerning the variation in the heating load with outside weather conditions as well as in the two systems of heating. The radiation room was heated by two radiators of equal size, placed under the two windows, with the heat supplied by bayonet heaters screwed into both bottom

connectors of each radiator. The electric heaters in each radiator were so arranged, with three-way switches, that one of the bayonet heaters could be supplied by hand control with any one of three constant rates of electrical input, while the heater in the other end of the radiator could be supplied with any one of three electrical input rates thermostatically controlled. This control provided a wide flexibility and, incidentally, a very uniform radiator surface temperature.

The convection room was equipped with an air circulating system installed in the space below the floor (convection room, Fig. 2). This system consisted

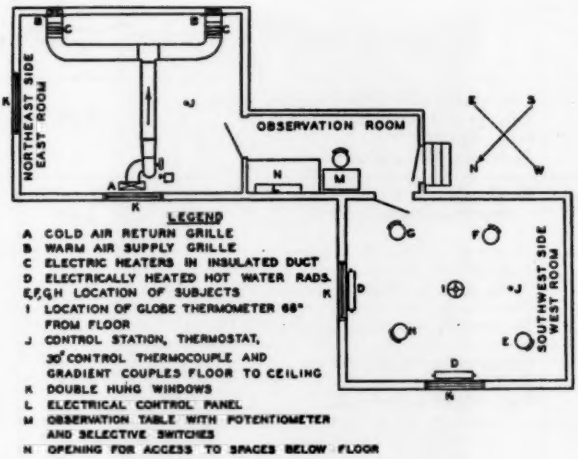
NORTHEAST SIDE
EAST ROOM
OBSERVATION ROOM
SOUTHWEST SIDE
WEST ROOM
LEGEND
A COLD AIR RETURN GRILLE
B WARM AIR SUPPLY GRILLE
C ELECTRIC HEATERS IN INSULATED DUCT
D ELECTRICALLY HEATED HOT WATER RADS.
E,F,G,H LOCATION OF SUBJECTS
I LOCATION OF GLOBE THERMOMETER 66" FROM FLOOR
J CONTROL STATION, THERMOSTAT, 30° CONTROL THERMOCOUPLE AND GRADIENT COUPLES FLOOR TO CEILING
K DOUBLE HUNG WINDOWS
L ELECTRICAL CONTROL PANEL
M OBSERVATION TABLE WITH POTENTIOMETER AND SELECTIVE SWITCHES
N OPENING FOR ACCESS TO SPACES BELOW FLOOR

FIG. 2. FLOOR PLAN OF TWO 12 FT X 15 FT TEST ROOMS AND OBSERVATION ROOM. DUPLICATE PROVISIONS IN BOTH ROOMS FOR RADIATORS AND AIR CIRCULATION SYSTEMS, LOCATION OF SUBJECTS, AND MEASUREMENT OF ALL ENVIRONMENTAL CONDITIONS

of a fan exhausting air from a single floor grille (6 in. x 20 in.) placed below the window in the northwest side of the room and returning this air to the room through two 6 in. x 12 in. deflecting type grilles, placed 1¼ in. from the floor and 18 in. from each corner, in the southeast wall of the room. Thermostatic control of the two-speed fan gave an air delivery of 57 cfm through each supply grille or 115 cfm to the room until the delivered air reached 110 F. When the air supply temperature exceeded this value the fan was turned on high speed, delivering twice these volumes until the temperature dropped to 100 F when the fan was again returned to the low speed. This arrangement gave a maximum outlet temperature of 107 F with the fan running at a high speed on the coldest day on which a test was made. Heating units were located in two insulated sections of the duct near the warm air supply grilles. Each of these two heaters was made up of identical electrical

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heating units connected in series so that equal rates of heat input were supplied to each grille. A portion of this heat controlled by three rate switches was under hand control, and an independently controlled rate was under thermostatic control.

In order to maintain the same relative humidity in both rooms and in any one room throughout a given test period, water-evaporator humidifiers, electrically heated and under humidistatic control, were employed. One was placed near the return grille in the convection room and the other in front of

TABLE 1—CONSTRUCTION CHARACTERISTICS OF TEST HOUSE

SURFACE	CONSTRUCTION	TRANSMISSION COEFFICIENT (U)	AREA IN EACH ROOM (SQ FT)
Outside Walls[a]	1-in. yellow pine siding building paper 1-in. yellow pine sheathing 6-in. studding ½-in. rigid insulation board	0.19	378
Windows (2 in each room)	Single glazed, double hung, wood sash, weatherstripped	1.13	35
Floors[b]	1-in. yellow pine ½-in. rigid insulation board on 2 x 6 in. joists	0.27	176
Ceilings	½-in. rigid insulation board on 2 x 4 in. rafters	0.46	176
Roofs	Composition strip shingles 30-lb felt 1-in. yellow pine sheathing on 2 x 4 in. rafters	0.53	184
Roof and Ceiling		0.25	176

[a] Test rooms only.
[b] Since the upper and lower surfaces of the floor were maintained at the same temperature to within ± 1 F heat flow through the floor was limited to ± 0.27 Btu per square foot per hour.

a radiator in the radiation room so that the moisture was drawn into the recirculated air stream before it was heated. All electrical energy supplied in either room, including that supplied to the heaters, humidifiers, psychrometry fans, and lights passed through calibrated watt-hour meters which were read hourly during all tests. All electrical circuits were controlled from a switchboard, Fig. 3, in the observation room with the exception of the choice of heat input rates to each end of the two radiators.

The 30-in. basement space between the roof of the warehouse building and the floor of the test rooms was independently heated by steam radiators (located below the floor level in the observation room). The air flow over each of these was thermostatically controlled so as to give a zero ± 1° F temperature differential between the lower and upper surfaces of the floor.

A precision potentiometer with selective switches installed in the observation room made it possible to observe the temperature at 27 points throughout either room as indicated in Fig. 4, giving the temperature of the wall surfaces on

the four sides of each room at seven points, the temperature of the air 6 in. away from five of these points, the temperature of the floor surface at one point, the temperature of the air 6 in. above this point, the temperature of the ceiling surface at two points, and the temperature of the air 24 in. below these two points. At the point indicated in Fig. 4 as the *control station* a ¼-in. pipe, flanged to the floor and ceiling, served to support the air thermocouples at the 6-, 30-, 78-, and 96-in. levels and the room thermostat at the 30-in. level.

Eight male college students between the ages of 19 and 22 years of age were selected as subjects for the study. These men were chosen as normal in every respect. They were given a thorough physical examination by a physician

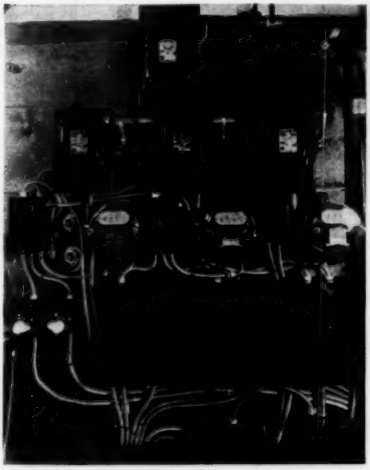

FIG. 3. VIEW SHOWING METERS AND SWITCHES USED IN RADIATION AND COMFORT TEST BUILDING

and only those whose metabolic rates, body temperatures, pulse rates, and other reactions which might have any thermal significance, were normal, were selected. These subjects wore normal attire for the winter season consisting of shorts; a sleeveless undershirt; shirt with collar and tie; socks; low cut shoes; and a medium weight, single-breasted suit including a vest, coat, and trousers. During a test they were seated in straight backed chairs and were allowed to engage in any sedentary activity which did not result in undue excitement or agitation, such as reading, preparing college lessons, or talking.

A globe thermometer for determining the mean radiant temperature (the integrated average temperature of all surroundings in view of a given point) was located in the center of each room at the 66-in. level. The sensitive element of a Pierce hot-wire anemometer was located in each test room near the globe thermometer. The globe thermometer temperatures and the anemometer observations were made from the observation room, thereby making it

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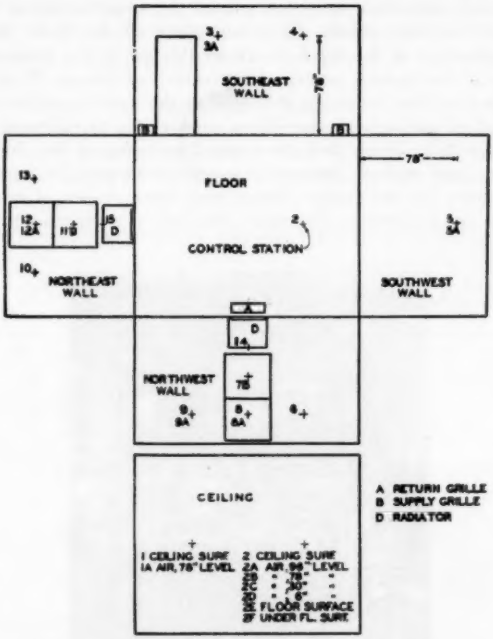

FIG. 4. LOCATION OF THERMOCOUPLES IN WEST ROOM. ARRANGEMENT IN EAST ROOM IDENTICAL WITH EXCEPTION OF DOOR

(+) with single number indicates location of thermocouples on surfaces. Numbers and letters locate thermocouples in air 6 in. from surface unless otherwise indicated. Indicated distances are from floor level. Thermocouples No. 2 located on ¼-in. pipe flanged to floor and ceiling at control station

unnecessary for any observer to come near any of these instruments. The characteristics of the globe thermometer, the auxiliary instruments used therewith and the determination of the mean radiant temperature will be discussed.

TEST OPERATIONS

In the operation of any test on a given day the temperature of the rooms was kept near the desired level throughout the previous night and was then placed on the thermostatically controlled electrical heaters in the early morning. With the beginning of the preliminary test period at about 1:00 p.m., the eight male subjects were seated somewhat symmetrically in each room at points *E*, *F*, *G*, and *H*, Fig. 2, so that no parts of their bodies came within 2 ft of

any wall surface or within 3 ft of any radiator surface. These four locations were further selected so that the subjects did not come within 3 ft of the control thermocouple, the thermostat or the entrance to the aspirating psychrometer. During the ¾-hour preliminary period, observations of the control temperature, a few of the wall surface temperatures, the wet- and dry-bulb temperatures and the globe thermometer temperatures were made.

At the close of the preliminary period the test proper was begun during which time the subjects remained in the room into which they originally entered for 80 min when they interchanged rooms for 80 min, and then again interchanged, so that each group of subjects was again in its original room during the last 80 min of the 4-hour test period. During this period the subjects indicated their feelings of warmth and any local sensations of draft, coolness, and excessive warmth at 15-min intervals.

After the first few preliminary tests it was found desirable to maintain a constantly changing effective temperature in each room rather than a constant condition. Thus, the temperature was either allowed to rise or to fall in both rooms simultaneously at a constant rate during the four hour period so that the subjects would slowly change from a condition where the average was slightly cool to a condition where the average was first comfortable and later slightly warm, or vice-versa. This was done in order to make available sufficient data so that, when plotted, they might indicate the particular time, and therefore the particular environmental conditions, at which the subjects were *ideally comfortable.*

During the 4-hour period frequent observations were made of all the temperatures, including the wet- and dry-bulb temperatures, globe thermometer temperature, both the exposed and the shielded air thermometer temperatures, and the air velocity (by the Pierce hot-wire anemometer).

TEST RESULTS

A log of one of the tests is given in Fig. 5 where the dry-bulb temperature, the wet-bulb temperature, and the resulting effective temperature at the 30-in. level *control station* are plotted for each room. An attempt was made to keep all three of these factors changing uniformly throughout the period of the test. Actually, the greatest attention was directed toward varying the effective temperature according to a predetermined curve and at the same time maintaining a constant relative humidity. This procedure necessitated the simultaneous control of the dry-bulb and the wet-bulb temperatures.

Also, in Fig. 5 is plotted for each room the mean radiant temperature as determined from the globe thermometer temperature, the globe temperature differential, the true air temperature, and the air velocity. A side wall temperature observed at a single point by thermocouple No. 5, Fig. 4, is also plotted.

In the lower part of the chart the average comfort votes for the four subjects, as observed at frequent intervals throughout the test, are plotted against the effective temperature of the conditions in each room at corresponding times. The comfort index here used was that employed in a number of earlier Laboratory studies during the past several years, in accordance with which a condition of *ideally comfortable* as far as the feeling of warmth is concerned is given an index number of *4*; the conditions of *comfortably cool*

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TABLE 2—DATA FROM 17 TESTS IN THE TWO ROOMS

TEST NO.	DATE	OUTSIDE AIR TEMPERATURE	SOLAR RADIATION B.T.U./4 HRS.	AVERAGE OUTSIDE AIR VELOCITY DURING TEST M./HR.	RADIATION ROOM	RADIATION ROOM: DRY BULB TEMP. 30" FROM FLOOR	78" FROM FLOOR	AVERAGE (30" and 78")	WET BULB TEMPERATURE	EFFECTIVE TEMPERATURE	DEGREES ET. CHANGE FOR 1 COMFORT INDEX CHANGE	MEAN RADIANT TEMP.	AVERAGE DB - MRT	AIR VELOCITY FT./MIN.	CEILING - FLOOR AIR TEMPERATURE DIFF.	HEAT INPUT: B.T.U./HR.	B.T.U./HR./°F. DIFFERENCE INSIDE - OUTSIDE	CONVECTION ROOM: DRY BULB TEMP. 30" FROM FLOOR	78" FROM FLOOR	AVERAGE (30" and 78")	WET BULB TEMPERATURE	EFFECTIVE TEMPERATURE	DEGREES ET. CHANGE FOR 1 COMFORT INDEX CHANGE	MEAN RADIANT TEMP.	AVERAGE DB - M.R.T.	AIR VELOCITY FT./MIN.	CEILING - FLOOR AIR TEMPERATURE DIFFERENCE	HEAT INPUT: B.T.U./HR.	B.T.U./HR./°F. DIFFERENCE INSIDE - OUTSIDE	HEAT INPUT ROOM R IN % ROOM C: % B.T.U./HR.	% B.T.U./HR./°F. DIFFERENCE INSIDE - OUTSIDE
A	B	C	D	E	F	G_R	H_R	I_R	J_R	K_R	L_R	M_R	N_R	O_R	P_R	Q_R	R_R	G_C	H_C	I_C	J_C	K_C	L_C	M_C	N_C	O_C	P_C	Q_C	R_C	S	T
14	1/11	36.3	0	9	W	73.3	74.3	73.8	60.1	68.5	2.0	72.3	1.5	30	2.1	17288.6	46.1	74.9	74.9	74.6	60.5	69.1	2.0	71.2	3.4	31	-0.1	14711	38.4	118	120
15	1/12	39.9	0	7	W	73.0	73.3	73.1	60.7	68.5	2.2	71.6	1.5	35	1.7	17460	52.6	73.4	73.4	73.4	61.0	68.7	2.2	71.4	2.0	31	0.1	15072	45.0	116	130
16	1/13	27.9	25	19	W	74.4	76.4	75.4	58.7	68.5	2.5	74.0	1.4	25	2.7	39745	83.7	75.0	76.5	75.8	59.7	68.0	2.3	71.7	4.1	31	1.7	33123	69.2	120	121
17	1/16	20.5	0	16	W	74.2	76.2	75.2	57.8	68.2	2.5	72.7	2.5	25	4.9	47819	87.4	76.2	78.2	77.2	57.8	69.2	2.9	72.9	4.3	30	2.2	44536	78.6	107	111
18	1/17	17.2	75	9	W	75.1	76.6	75.8	57.6	68.6	3.7	74.6	1.2	21	2.3	23411.7	39.9	77.5	78.8	78.1	58.1	68.9	3.4	73.3	4.8	25	2.0	29737	48.8	79	83
19	1/19	13.0	400	14	W	75.0	75.7	75.3	57.2	68.3	3.1	74.4	0.9	20	3.3	39149	62.8	78.0	79.4	78.7	58.7	70.3	3.3	74.0	4.7	45	2.5	50261	76.5	78	82
20	1/19	4.7	0	14	W	73.1	77.1	76.1	59.0	68.8	2.9	74.4	1.7	23	3.5	77369	109.2	77.0	78.0	77.5	60.5	70.0	2.3	70.3	7.2	32	2.2	68214	93.7	114	118
22	1/23	26.5	420	9	W	75.2	77.1	76.1	58.1	68.9	3.6	75.2	0.9	27	2.6	45905	92.6	76.3	76.7	76.5	59.0	69.6	4.1	71.0	5.6	40	1.7	35515	71.0	129	130
23	1/24	27.5	0	14	W	74.8	76.4	75.6	58.1	68.6	2.4	73.2	2.4	25	2.0	39668.5	115.8	75.5	75.0	75.1	59.1	69.1	2.2	70.0	5.1	40	3.0	40474	85.1	138	136
24	2/1	33.9	69	12	E	75.1	77.6	76.3	58.2	68.7	2.3	74.0	2.3	22	3.0	49068	115.8	76.6	78.6	77.6	60.0	69.9	2.5	74.1	3.5	45	3.0	45972	105.2	107	110
25	2/2	17.0	457	16	E	75.0	76.4	75.7	58.0	68.5	2.3	74.5	1.2	25	0.9	60942	109.8	76.0	78.9	77.4	58.3	69.3	2.5	74.0	3.4	32	2.3	59198	98.0	103	106
26	2/3	38.6	42	5	E	73.2	75.7	74.4	60.9	68.5	1.9	72.2	2.2	26	3.0	34722	97.0	73.3	73.9	73.6	60.5	68.3	3.7	70.6	3.0	35	2.1	31794	90.8	109	107
27	2/7	32.0	0	10	E	72.7	73.7	73.2	60.0	68.0	3.4	69.4	3.3	46	3.2	45444	110.3	73.6	74.3	73.9	61.0	68.8	2.9	71.2	2.7	30	1.8	42068	100.4	108	110
28	2/8	40.2	309	6	E	73.5	73.4	73.4	60.7	68.5	2.8	72.2	1.2	25	0.5	16666	50.2	72.8	73.2	73.5	61.6	68.8	1.7	71.4	1.1	26	0.8	21618	64.9	77	75
30	2/13	32.3	0	6	E	74.0	74.2	74.1	61.6	69.1	3.5	70.4	3.7	25	3.1	47677	114.1	75.2	74.9	75.0	62.9	70.1	3.5	71.6	3.4	30	1.2	34893	86.4	12.9	132
31	2/14	26.6	197	23	E	74.6	74.2	74.4	59.2	68.7	2.0	72.0	2.4	25	1.3	31342	65.6	75.6	75.6	75.6	62.6	69.7	2.5	72.2	3.4	41	1.2	29645	60.5	106	109
32	2/15	30.8	1106	13	E	75.0	75.1	75.0	59.3	69.0	2.7	73.5	1.5	26	0.2	13026	29.5	75.5	75.5	75.3	60.2	69.5	2.9	73.1	2.2	32	1.1	15806	35.5	83	82
AVERAGE VALUES						74.3	75.5	74.9	59.2	68.6	2.7	73.0	1.9	27	2.4	39026	80.9	75.4	76.2	75.8	60.1	69.4	2.7	72.0	3.8	33.6	1.7	36155	73.5	108	110

In Tests Nos. 18 and 19 windows in radiation room were blanked off with 1 in. structural board insulation.

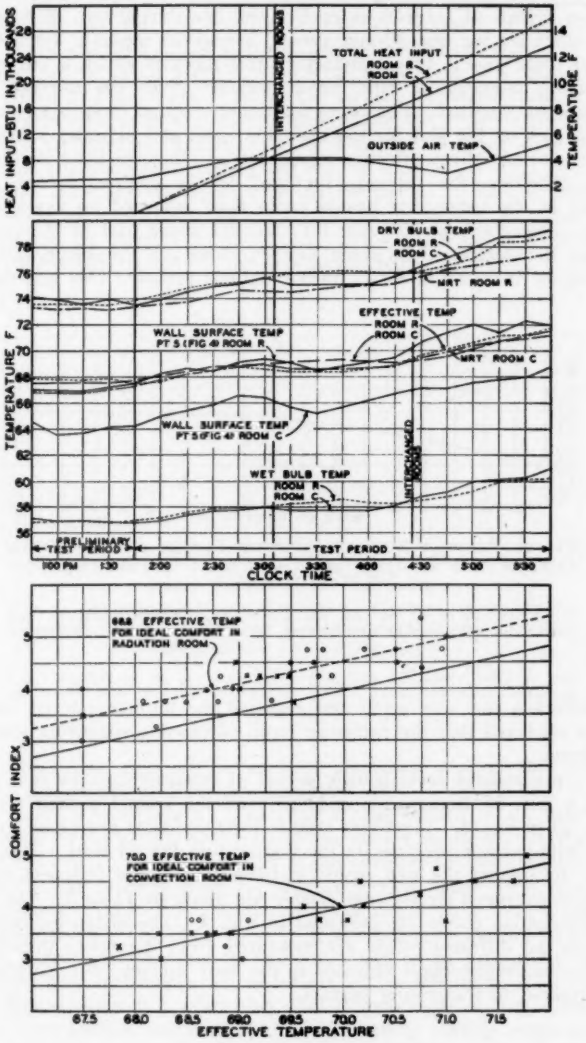

FIG. 5. LOG OF TEST NO. 20 ON JANUARY 19, 1940 (R—RADIATION ROOM) (C—CONVECTION ROOM)

and *comfortably warm* are given index numbers of *3* and *5*, respectively; *cool* and *warm*, *2* and *6*, respectively; and *cold* and *hot*, *1* and *7*, respectively.

The two groups of subjects in each room, one during two 80-min periods, and the other during one 80-min period, are given separate symbols. It is noted that these give a very satisfactory plot through which a curve may be drawn whose intersection with the *4* ordinate gives the average effective temperature for comfort. The curve for the convection room is super-imposed for comparison on the plot for the radiation room and indicates a lower index of warmth for any given effective temperature. Thus, in the radiation heated room on this day, the subjects averaged *ideally comfortable* at 68.8 deg ET, while in the convection heated room they indicated an average feeling of *ideally comfortable* at 70.0 deg ET. In the upper part of the chart there are plotted the electrical heat input to each room and the outside air temperature, ranging from 2.4 to 5.2 F, during the test period. This was the coldest

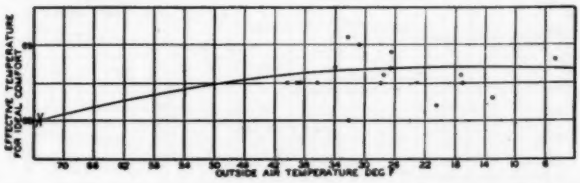

FIG. 6. RELATION BETWEEN EFFECTIVE TEMPERATURE AT WHICH COMFORT WAS REALIZED AND CONCURRENT OUTSIDE AIR TEMPERATURE FOR RADIATION ROOM

(×) dry-bulb at 40 per cent RH and 68 deg ET where subjects were comfortable with no difference between air and mean radiant temperatures

day on which a test was made and required the highest rate of heat input. It will be observed that the radiation room required a somewhat greater rate of heat input.

After a preliminary investigating period of research into the best means of handling a test, 17 tests were conducted and the results plotted similarly to Test No. 20 in Fig. 5. Important results for these tests are given in Table 2, together with certain averages in the bottom line for all tests. Temperatures observed throughout the two rooms at times during the different tests when the subjects averaged *ideally comfortable* are given in Table 3. The locations of the observation points are indicated in Fig. 4. The distances, expressed in inches, for the different points are measured from the floor; therefore, zero represents the thermocouple attached to the floor surface; 102 in., the thermocouple attached to the ceiling surface.

ANALYSES OF TEST RESULTS

A comparison of the test results from the radiation and convection rooms indicates a universally lower effective temperature for comfort in the radiation room than for the convection room. This is accounted for by the warming

effect on the subjects by direct radiation from the radiators and that reflected and reradiated from other surfaces within the room. The average values in Table 2 indicate that the average temperature for comfort for all tests in the radiation room was 68.6 deg ET, while for the convection room the effective temperature had to be an average of 0.8 deg higher, or 69.4 deg ET. This is better shown in Figs. 6 and 7, where the outside air temperature at the time when the subjects were *ideally comfortable* within the room is plotted against the effective temperature required for ideal comfort. Tests were conducted when outside temperatures ranged from 4 F to 40 F.

The points *X*, fixing the upper ends of the curves at 73.6 F outside temperature, were determined independently in the psychrometric rooms of the Laboratory for the purpose of this study. This procedure will be discussed later. It represents an equilibrium condition where the outside dry-bulb, the

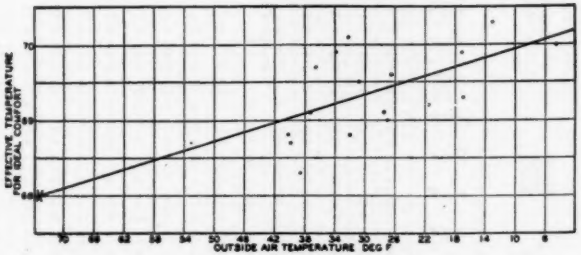

FIG. 7. RELATION BETWEEN EFFECTIVE TEMPERATURE AT WHICH COMFORT WAS REALIZED AND CONCURRENT OUTSIDE AIR TEMPERATURE FOR CONVECTION ROOM

(×) dry-bulb at 40 per cent RH and 68 deg ET where subjects were comfortable with no difference between air and mean radiant temperatures

inside dry-bulb, and the wall surface temperatures were all the same with 68 deg ET and 40 per cent relative humidity.

While there is considerable scattering of the points, these tests indicate definite trends in the results as shown by the curves as drawn. For the radiation room there is very little variation in the effective temperature for comfort given by the test results, while for the convection room there is a definite trend, indicating a desire for a higher effective temperature with a drop in the outside temperature. This may logically be accounted for by the fact that with decreasing outside temperature the radiators had to be maintained at higher temperatures with increasing radiant effects within the room. It appears that in the radiation room this effect just about compensates for the lowering of the inside wall surface temperature with decreasing outside temperature. In the convection heated room where there was no compensating radiant effect with decrease in outside temperature, the cooling effect with decrease in outside temperature; that is, the cooling effect of radiation from occupants to the colder walls, required a compensation in the form of higher dry-bulb and effective temperatures.

XUM

TABLE 3—TEMPERATURES OBSERVED THROUGHOUT THE TWO ROOMS AT A TIME DURING THE DIFFERENT TESTS WHEN THE SUBJECTS AVERAGED COMFORTABLE

	Test Number	Date	Outside Air	Time of Observation	Ceiling			Central Control					Southeast Wall			Southwest Wall		Northwest Wall						Northeast Wall					
					1 102″ X	1A 78″ Z	2 102″ X	2A 96″ V	2B 78″ Z	2C 30″	2D 6″ V	2E 0″ X	3 78″ X	3A 78″ V	4 78″ X	5 78″ X	5A 78″ V	6 78″ X	7 48″ G	8 78″ G	8A 78″ VG	9 78″ X	9A 78″ V	14 24″ R	10 78″ X	11 48″ G	12 78″ G	12A 78″ VG	13 78″ X	15 24″ R
Room R — West Room	14	1-11	36.3	3:30	69.5	74.3	68.2	74.3	74.3	73.3	73.2	72.6	70.1	74.1	68.5	68.7	73.8	69.0	55.6	55.8	74.7	69.5	74.3	96.4	68.7	58.5	56.6	75.0	69.3	96.7
	15	1-12	39.9	2:45	69.1	73.4	68.3	74.1	73.3	73.0	72.4	71.8	69.4	73.3	69.3	68.5	73.2	69.1	59.4	57.7	73.9	69.2	73.4	99.4	68.8	58.5	57.0	74.1	69.0	100.1
	16	1-15	27.9	3:30	70.8	76.3	68.9	76.4	76.4	74.4	73.7	72.0	69.9	76.3	69.7	68.9	75.0	69.7	49.9	48.4	78.6	69.1	75.9	120.2	68.6	51.9	50.5	78.6	69.2	121.7
	17	1-16	20.5	2:09	70.2	76.4	67.9	77.6	76.2	74.2	72.7	70.0	69.3	76.9	67.1	67.5	76.4	67.2	47.7	45.8	83.0	68.2	77.4	131.3	67.0	47.7	50.1	83.1	67.7	133.0
	18	1-17	17.2	4:33	71.9	76.7	70.3	77.4	76.6	75.1	75.1	75.5	71.8	76.7	69.7	70.1	76.6	71.2	Window Blanked Off			69.9	76.3	93.8	68.9	Window Blanked off			73.3	85.1
	19	1-18	13.0	5:30	69.7	77.6	71.1	77.0	75.7	75.0	73.7	73.5	70.4	77.2	69.0	68.5	76.1	69.4				68.0	76.2	127.1	68.0				71.7	93.4
	20	1-19	4.7	3:15	70.0	78.1	69.2	77.6	77.1	75.1	74.1	72.4	70.4	77.6	68.6	67.8	77.1	67.4	39.4	37.6	79.3	67.6	77.2	137.2	65.9	38.5	41.3	79.5	66.2	136.6
	22	1-23	26.5	2:45	73.6	78.7	70.3	78.6	77.1	75.2	75.6	75.3	73.1	78.2	70.5	71.5	78.1	70.6	59.0	56.2	83.3	70.3	78.6	124.7	70.0	56.0	54.9	84.0	70.6	122.0
	23	1-24	27.5	3:48	71.1	76.9	69.5	76.7	76.4	74.8	73.7	70.6	69.8	78.3	70.8	69.3	77.0	71.0	51.8	50.8	82.6	71.7	78.6	127.5	68.5	52.0	51.8	82.6	72.2	129.7
Room R — East Room	24	2-1	33.9	3:15	70.5	75.4	72.4	77.0	77.6	75.1	74.0	74.1	69.9	74.9	71.1	69.6	76.4	70.3	57.6	55.8	85.6	69.4	74.8	147.0	71.8	50.8	52.7	74.5	69.7	81.8
	25	2-2	17.0	3:45	70.7	76.1	70.4	74.8	76.4	75.0	74.9	74.1	68.7	75.8	70.9	71.1	74.9	68.3	45.1	44.4	79.6	69.0	76.1	128.7	67.6	49.8	48.9	78.8	67.9	126.6
	26	2-5	38.6	4:30	68.6	75.4	70.9	75.8	75.7	73.2	72.5	72.0	69.0	74.7	68.2	68.3	73.8	69.0	56.8	55.6	77.5	69.5	74.9	118.4	68.3	59.8	59.8	78.5	68.5	115.1
	27	2-7	32.0	3:18	67.7	73.6	68.3	75.5	73.7	72.7	72.3	72.0	66.5	74.0	67.9	66.7	73.1	71.3	53.1	50.7	75.9	67.5	74.6	111.3	66.9	54.4	53.9	76.0	67.3	109.8
	28	2-8	40.2	4:15	70.5	74.3	70.1	73.6	73.4	73.5	73.1	73.0	68.6	73.4	69.2	69.9	73.5	68.8	55.6	54.8	72.5	68.3	73.6	95.1	68.7	54.3	54.7	73.6	69.0	94.1
	30	2-13	32.3	4:13	68.3	74.4	69.8	76.2	74.2	74.0	73.1	72.6	67.7	74.8	68.7	69.3	74.0	71.0	56.4	55.5	77.3	68.2	74.5	118.0	67.0	54.5	55.5	77.4	67.0	117.4
	31	2-14	26.6	4:26	69.6	75.8	70.6	75.2	74.2	74.6	74.2	73.6	68.8	75.3	70.8	69.4	74.7	72.0	51.1	48.8	76.3	69.3	75.3	112.3	68.2	51.1	50.0	77.5	68.4	110.8
	32	2-15	30.8	3:45	69.5	74.7	69.4	75.1	75.1	75.0	74.9	74.8	71.6	75.1	71.6	73.6	75.0	69.4	50.4	50.2	74.4	69.1	74.7	97.6	68.4	51.5	52.7	74.1	68.4	99.2

ROOM C — EAST ROOM	14	1–11	36.3	3:30	69.5	74.6	70.9	74.6	74.9	74.3	74.7	77.0	69.7	74.7	70.0	69.8	74.2	71.0	57.9	58.1	73.0	69.6	74.2	.....	69.4	56.0	56.4	72.6	69.4	.....
	15	1–12	39.9	2:30	69.2	73.3	69.3	74.3	73.4	73.4	74.2	74.5	69.5	73.5	69.8	69.3	74.5	70.4	56.6	56.2	73.2	70.3	73.9	.....	69.3	56.5	56.5	73.1	69.5	.....
	16	1–15	27.9	3:30	71.7	74.2	72.0	75.9	76.5	75.0	74.2	75.7	68.5	74.8	70.0	70.0	75.8	69.0	45.6	48.2	73.4	67.6	73.9	.....	67.5	46.4	47.6	72.6	68.4	.....
	17	1–16	20.5	4:00	70.6	79.6	72.8	78.4	78.2	76.2	76.2	77.3	70.1	77.0	71.7	71.3	77.3	71.2	53.3	55.4	76.7	70.6	79.2	.....	70.3	48.6	49.1	77.1	70.4	.....
	18	1–17	17.2	4:30	70.9	77.7	71.2	78.7	78.8	77.3	76.7	76.5	68.6	76.9	69.9	71.5	77.7	70.4	45.7	50.8	77.1	69.8	79.3	.....	67.7	47.0	47.5	75.8	68.4	.....
	19	1–18	13.0	3:15	70.1	80.3	73.7	81.2	79.4	78.0	78.7	77.8	70.8	80.7	73.9	72.1	80.1	72.3	42.9	43.3	77.4	70.9	79.9	.....	70.0	44.8	46.4	78.0	69.9	.....
	20	1–19	4.7	2:24	69.6	77.6	70.0	78.0	78.0	77.0	75.8	74.2	67.0	77.5	69.4	67.9	77.3	69.8	33.6	35.8	76.1	66.4	74.3	.....	65.1	38.6	40.7	74.6	65.9	.....
	22	1–23	26.5	3:15	68.2	75.2	71.4	75.7	76.7	76.3	75.3	73.8	68.7	74.8	70.2	70.0	76.8	69.5	49.4	50.1	74.2	68.8	74.9	.....	64.8	46.6	49.4	75.0	63.8	.....
	23	1–24	27.5	3:05	69.7	74.8	70.7	75.9	75.0	75.3	72.8	70.7	65.7	74.3	67.2	67.5	75.7	67.1	45.1	45.8	75.4	66.7	75.4	.....	65.9	46.1	48.7	73.4	66.2	.....
ROOM C — WEST ROOM	24	2–1	33.9	2:45	72.5	78.4	72.2	78.2	78.6	76.6	73.7	71.4	71.7	77.2	71.0	70.8	77.9	72.5	50.4	51.1	76.3	70.6	77.9	.....	69.7	51.8	50.2	77.2	76.4	.....
	25	2–2	17.0	4:03	71.8	79.2	72.5	79.8	78.9	76.0	73.6	72.0	72.9	80.3	71.2	71.4	78.8	68.3	42.3	42.4	78.0	69.8	78.6	.....	69.2	42.6	42.8	78.1	72.6	.....
	26	2–5	38.6	3:11	69.5	73.8	69.3	74.4	73.9	73.3	71.3	70.6	69.6	74.8	68.0	69.2	74.3	68.6	50.6	51.6	73.2	68.5	73.6	.....	67.8	53.1	53.7	73.3	70.7	.....
	27	2–7	32.0	3:50	69.1	75.0	69.5	75.2	74.3	73.6	71.9	70.8	69.1	75.5	68.4	68.6	73.9	67.5	47.8	48.1	74.2	67.8	73.6	.....	67.2	49.7	51.6	74.5	71.2	.....
	28	2–8	40.2	4:51	68.3	71.6	70.5	71.9	72.2	72.8	72.1	72.2	69.3	71.9	68.3	69.8	72.6	67.9	54.5	54.4	72.7	68.2	72.6	.....	67.4	54.5	55.4	70.9	68.0	.....
	30	2–13	32.3	4:45	68.1	75.1	69.1	76.0	74.9	75.2	74.0	71.4	68.8	74.6	67.7	69.2	74.9	71.4	51.4	49.7	72.4	67.2	71.1	.....	67.7	49.0	49.8	74.3	72.3	.....
	31	2–14	26.6	3:24	69.4	75.8	71.5	72.9	75.6	75.6	74.5	73.9	70.1	76.5	69.4	71.5	76.5	69.5	50.0	51.9	75.8	70.7	76.0	.....	68.8	48.8	50.9	75.6	73.8	.....
	32	2–15	30.8	3:40	70.9	75.2	69.8	75.6	75.5	75.5	74.0	73.9	72.6	74.7	71.3	73.4	76.0	69.6	49.5	50.3	73.0	68.8	73.5	.....	68.3	50.2	50.3	73.2	62.8	.....

Location of observation points as indicated in Fig. 4. Distances measured from floor.

Therefore:

0 in. = floor surface
102 in. = ceiling surface
X = surface temperatures
Y = air temperature 6 in. from surfaces
Z = air temperature 24 in. from surfaces
G = Glass surface temperature
R = radiator surface temperature

While the variation of the required indoor effective temperature with outdoor dry-bulb temperature in the two rooms is significant, the exact magnitude, as indicated in Figs. 6 and 7, will naturally vary with the insulating value of the walls of the enclosure, and the percentage of the walls exposed to the outside.

The variations in outside dry-bulb temperature with the mean radiant temperature, the difference between the average dry-bulb temperature within the room and the mean radiant temperature, and the average indoor dry-bulb temperature are all plotted for the radiation and convection rooms in Figs. 8 and 9, respectively. Again, while there is considerable variation in the points so that exact curves cannot very satisfactorily be drawn, definite tenden-

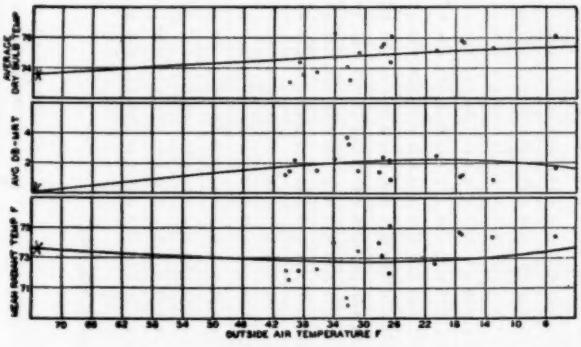

FIG. 8. RELATION BETWEEN OUTSIDE DRY-BULB TEMPERATURE AND MEAN RADIANT TEMPERATURE, AVERAGE DRY-BULB MEAN RADIANT TEMPERATURE DIFFERENCE AND AVERAGE DRY-BULB TEMPERATURE (30-IN. AND 78-IN. LEVELS) FOR RADIATION ROOM

(×) dry-bulb at 40 per cent RH and 68 deg ET where subjects were comfortable with no difference between air and mean radiant temperatures

cies are indicated which differ widely for the two rooms. In the convection room there is a definite increase in the difference between the average air temperature and the mean radiant temperature, with lower outside air temperature, while the same difference for the radiation room is either constant, or possibly there is some slight tendency for a decrease in this temperature difference with lower outside temperature. This indicates again that the increased radiation from the warmer radiator in colder weather compensates for the increased radiation loss from the subject to the colder walls and windows, thus accounting for the demand for increasing effective temperature to maintain comfort in the convection room with little or no increase in this demand for the radiation room.

The mean radiant temperature observed in the two rooms is plotted against the effective temperature for ideal comfort in Fig. 10. No relation is shown. Two distinct factors serve to confuse the relationship which might be expected to exist; that is, provided all other environmental factors remain constant, the

higher the mean radiant temperature the lower the required effective temperature. These two disturbing factors are: first, the fact that the dry-bulb temperature of the air for a given effective temperature, and therefore indirectly to a lesser extent the mean radiant temperature, increases with decreased moisture content of the air; and second, the fact that there was a small though persistent average greater air movement in the convection heated room than was observed for the radiation room.

The relation between effective temperature for ideal comfort and the difference between the average of the dry-bulb temperatures at the 30- and 78-in. levels, and the mean radiant temperature is plotted in Fig. 11 for both rooms. A rather definite curve is indicated, following the logically expected trend,

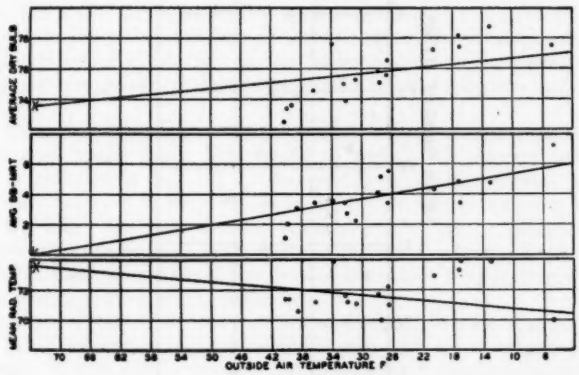

FIG. 9. RELATION BETWEEN OUTSIDE DRY-BULB TEMPERATURE AND MEAN RADIANT TEMPERATURE, AVERAGE DRY-BULB MEAN RADIANT TEMPERATURE DIFFERENCE AND AVERAGE DRY-BULB TEMPERATURE (30-IN. AND 78-IN. LEVELS) FOR CONVECTION ROOM

(×) dry-bulb at 40 per cent RH and 68 deg ET where subjects were comfortable with no difference between air and mean radiant temperatures

namely, that with increasing difference between the dry-bulb temperature of the air and the mean radiant temperature, a higher effective temperature is required for comfort. The higher mean radiant temperature and therefore the lower the dry-bulb to mean radiant temperature difference in the radiation room is shown by the segregation between the points for the radiation room and those for the convection room.

SENSATIONS OF DRAFT, COOLNESS, AND WARMTH

In stating their sense reactions to the conditions in the rooms the subjects were instructed to consider their feelings as regarding *first*, drafts, *second*, local sensations of coolness (if they did not interpret them as drafts), *third*, local sensations of warmth and *fourth*, any other unusual reactions they might have to the conditions in the room. Two of the subjects either took this too

seriously or were unusually sensitive for almost without exception they reported one or more of these unusual sensations. Excluding these two subjects there were seven sensations of draft for 149 *subject-periods* in the radiation room at times when the average subjects indicated feelings of *ideally comfortable*

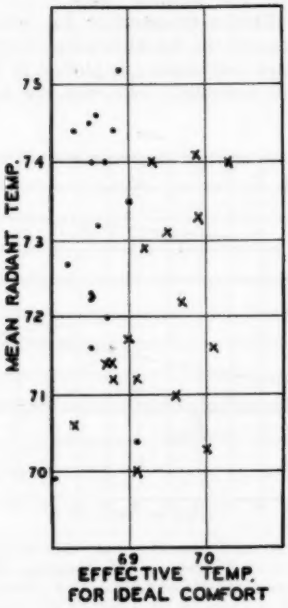

FIG. 10. RELATION BETWEEN EFFECTIVE TEMPERATURE AT WHICH COMFORT WAS REALIZED AND MEAN RADIANT TEMPERATURE

(·) indicates subjects in radiation room. (×) indicates subjects in convection room

and 11 sensations of draft for 148 *subject-periods* in the convection room. With the same *subject-periods* there were no local sensations of coldness and only one sensation of excessive warmth in the radiation room, and eight sensations of local coldness, and three sensations of excessive warmth in the convection room.

The two unusual subjects reported 19 local sensations of draft, one of coolness, and none of warmth for 50 *subject-periods* in the radiation room, and 71 local sensations of draft, 36 of coolness, and 11 of warmth in the convection

room for 49 *subject-periods*. Inasmuch as the subjects changed the positions in which they sat from test to test, it is not likely that the unusual difference in number of local sensation votes can be attributed to differences in position.

VARIATION IN THE EFFECTIVE TEMPERATURE REQUIREMENTS

The rather high effective temperature, demanded by the subjects for comfort, even on relatively warm days, was somewhat surprising. For example, it will be seen from the curves Figs. 6 and 7, that for outside temperatures in the neighborhood of 40 F, approximately 68.6 deg ET was required in the radiation room, and 69.0 deg ET in the convection room. These requirements did not seem to correlate well with the generally accepted 66 deg ET for comfort in warm wall rooms used currently in THE ASHVE GUIDE 1940. In order to give additional information on this point and in order to give directly comparable requirements for comfort in warm rooms, three extra tests with the same subjects were conducted in the Laboratory's psychrometric chambers following the completion of the tests reported for the radiation and convection rooms.

These three extra tests were similar to those conducted and reported [1] by the Laboratory at earlier dates; but instead of finding the average subject comfortable at from 64-66 deg ET, as observed in the earlier studies, these eight subjects were found to be comfortable in the psychrometric rooms at an average of 67.5 deg ET. The only obvious reasons which may be advanced for this change are either individual differences in the eight subjects from those used in the earlier studies, or a change in the desired indoor temperature for comfort since the earlier studies were conducted.

In the three extra tests made in the psychrometric rooms great care was taken to bring the wall surfaces into equilibrium with the atmospheric condition as had been done in earlier studies. In these tests there were available means for observing the mean radiant temperature which were not had in the earlier investigations. The actual mean radiant temperature found in the psychrometric rooms was about 1½ F above the air dry-bulb temperature at the 30-in. level. This apparently resulted from radiation from the bodies of subjects and observers and from the lighting. While no tests were made, it is probable that under such equilibrium conditions with no subjects or lights in the room there would have been no difference between the air temperature and the mean radiant temperature of the surface.

RADIATION AND ITS MEASUREMENT

Since 1920 the ASHVE Laboratory and others in this country have been studying the effect of the temperature, moisture content, and movement of the air upon a person's feeling of warmth. During the same period contemporaries in England led by Leonard Hill, Vernon, and Bedford, have paid a great deal of attention to the effect of the fourth environmental factor, radiation. Perhaps the principal reason for this difference in interest has been the greater attention paid in England to the environmental effects on industrial workers exposed to high temperatures where radiation is of great importance.

The studies abroad have naturally resulted in consideration of methods of

[1] See Bibliography 4, 5, 6, and 8.

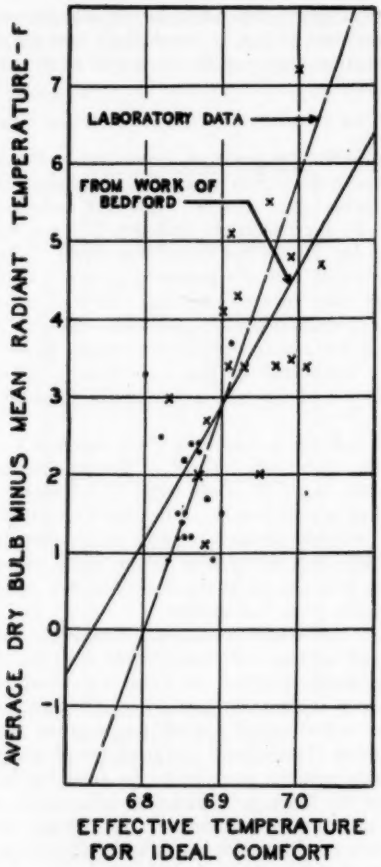

FIG. 11. RELATION BETWEEN EFFECTIVE TEMPERATURE AT WHICH COMFORT WAS REALIZED AND DIFFERENCE BETWEEN DRY-BULB AND MEAN RADIANT TEMPERATURE

(·) indicates subjects in radiation room. (×) indicates subjects in convection room. Solid curve shows this relationship based on work of Bedford and the early work of the Laboratory

measuring and evaluating radiation and the development of the Vernon globe thermometer, which is an instrument designed to give an integrated measure of the radiation effect at a point from the entire 4π radians of solid angle

surrounding it. Other instruments used in the past few years in this country and abroad are the eupatheoscope and the thermo-integrator.[2] The Vernon globe thermometer consists of a thin-shelled copper sphere 6 in. in diameter, blackened on the outside, with the sensitive bulb of a mercury thermometer inserted into the center. This globe is sensitive to radiation, air temperature, and air movement. If it is placed within an enclosure where the inside wall surfaces are above the inside air temperature, the radiation received by the sphere will raise its temperature to a point above that of the surrounding air. This difference between the air temperature and the globe temperature is a function of the radiation received from all surfaces in view of the globe. However, in order to be used to evaluate the magnitude of the radiation effect, a correction must be made for the air velocity near the globe in order to correct for the reduction in the resulting temperature difference.

Thus the Vernon globe thermometer requires not only the temperature of the globe itself but the temperature of the surrounding air unaffected by radiation, determined with a thermometer shielded with bright metal from radiation, and the air velocity near the globe determined by the Kata thermometer.

The use of this instrument at best demands unusual skill. *First,* its blackened surface must approach very nearly that of a true black body. *Second,* the determination of true air temperature unaffected by radiation is difficult and the ordinary method of shielding against radiation by bright metal cylinders is questionable. *Third,* the determination of small air velocities in occupied spaces by the Kata thermometer is a difficult and laborious task.

One inherent difficulty in the use of the Vernon globe thermometer results from the fact that the observer must come in close proximity to the instrument in order to determine the globe temperature, the shielded air temperature near it, and the Kata thermometer observations, thus unavoidably disturbing the very environmental conditions being measured.

The Laboratory has been able to improve the globe thermometer in its adaptability to the problem at hand by applying thermocouples not only for observing the globe temperature but also for observing the air temperature unaffected by radiation. It has been a well accepted fact that a very fine, bright metal thermocouple with no bead at the junction probably gives as nearly the true air temperature unaffected by radiation as is possible by any available methods. Based upon this fact the Laboratory developed an instrument in which a differential thermocouple of No. 36 copper-constantan wire was used, with its eight junctions alternately placed within the globe and in the air at its four sides; thus giving directly and more sensitively the temperature difference between the globe and its surrounding air. The use of one of these junctions as a single couple served to give the true globe temperature and that of another the true air temperature. This arrangement had the added advantage that observations could be made by means of the potentiometer located in the observation room. The use of a sensitive hot-wire anemometer developed by the John B. Pierce Laboratory made it possible to read the surrounding air velocity remotely; that is, its sensitive element was placed near the globe and read in the observation room.

The importance of the factors in the use of this instrument, previously described, is indicated by the fact that 100 per cent differences in resultant

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[2] See Bibliography 12.

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observations were had at times between the standard Vernon globe and the Laboratory instrument. The greater variability in the results in the older instrument together with the fact that the improved instrument always gave the greater resultant effect (any errors due to imperfection in black body surface, and due to the effect of radiation on the true air temperature would tend to make the reading too small) indicates that these improvements resulted in more accurate evaluations. Such possible variations indicate clearly that further study should be given to desirable instruments and methods for their use in observing mean radiant temperatures.

Fig. 12 shows the standard Vernon globe thermometer with the shielded thermometer, Kata thermometer, and bare air thermometer at the reader's

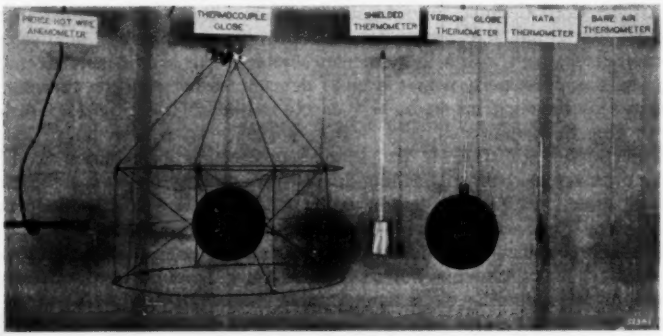

FIG. 12. VIEW OF INSTRUMENTS FOR MEASURING AIR ENVIRONMENT

right, and the instrument developed by the Laboratory with the Pierce hot-wire anemometer at the left.

British usage has developed the term *mean radiant temperature,* which is assumed to be the integrated average temperature of all surrounding surfaces from which radiation may come. Otherwise defined, mean radiant temperature is meant to be that uniform temperature to which all surfaces in view of a point must be heated in order to give the same resultant effect.

While the mean radiant temperature thus defined and measured by the globe thermometer is the best expression of the effect of radiation at a point available at the present time, there are criticisms of its use in defining the environmental effect of radiation for the reason that a person is sensitive to the distribution of radiation from various directions, as well as to the total effect. As an example, a person would undoubtedly feel very different in the following two conditions, both of which might give the same mean radiant temperature; *first,* a condition where the given mean radiant temperature is the result of all the surfaces in view of the point under consideration being at one temperature; and *second,* a condition where the same mean radiant temperature resulted from a very high temperature source in one direction (such as a large steam-heated radiator), and a low temperature source in the other direction (such as

a large single-glazed window on a sub-zero day). It is probable that a directional instrument consisting of a single radiation measuring cone which might be oriented at will, or a single instrument having six equal segments, taking in practically the entire surrounding solid angle, would serve admirably. While the combined effect given by the six segments of this instrument would neither give directly the mean radiant temperature, nor as complete an integration of the combined enclosure as the globe, it would probably be sufficiently accurate for the purpose, and would at the same time give the individual directional effects which are probably of greater value in our air conditioning practice than are the over-all effects, particularly in this country where occupants of air conditioned spaces seem to be sensitive to directional feelings of warmth or coldness.

A little attempt was made in this study to measure these directional effects by determining the contribution of each surface in the room to the mean radiant temperature as measured by the globe thermometer. A test was conducted in the rooms without occupants when the mean radiant temperature was measured by the globe thermometer, and the temperatures of all surfaces in the rooms were measured by both thermocouples and a cone-shaped radiation meter. The results of this test are given in Table 4. For the radiation room Column A gives the surface under consideration: B, the solid angle, expressed in radians, subtended by the surface from a point at the center of the Vernon globe thermometer; C, the temperature of the surface in question as observed by thermocouples; and D, this same temperature expressed in degrees absolute and raised to the fourth power. Column E gives the product of the solid angle and the surface temperature; and F, the product of the solid angle times the surface temperature expressed in degrees absolute and raised to the fourth power. Columns C′, D′, E′, and F′ give similar values observed in the convection room.

The solid angles, Column B, were evaluated in radians and their sum should equal the entire 4π radians in the solid spherical angle. Actually, the integration gave 12.563 or 3.99π radians. While the mean radiant temperature in the radiation room as observed by the globe thermometer was 69.3 F, the integrated average surface temperature resulting from summing up Column E was 65.4 F. The calculated mean radiant temperature based upon the fourth power of the temperatures was 67.9 F. For the convection room a similar integration is indicated. It is worthy to note that in the convection room where the surfaces were at more nearly the same temperature there is no appreciable difference between the calculated mean radiant temperature and the calculated integrated average temperature; while in the radiation room the presence of radiators approximately 60 F hotter than the integrated average temperature raised the calculated mean radiant temperature 2.5 F above the calculated average temperature.

EFFECTIVE TEMPERATURE—MEAN RADIANT TEMPERATURE RELATIONSHIP

It should naturally be assumed that a certain definite relationship should exist between the comfort of occupants in an enclosure and the mean radiant temperature, provided all other environmental factors remain unchanged. Closer correlation between the work of the Laboratory and our English contemporaries

TABLE 4—RELATION BETWEEN MEAN RADIANT TEMPERATURE DETERMINED BY GLOBE THERMOMETER AND CALCULATED FROM SURFACE TEMPERATURES AND SOLID ANGLES SUBTENDED BY THESE SURFACES

A	B	Radiation Room				Convection Room			
		C	D	E	F	C′	D′	E′	F′
Surface	Solid Angle Radians	Surf. Temp.		B x C	B x D	Surf. Temp.		B x C′	B x D′
		°F	T⁴			°F	T⁴		
S. E. Wall....	2.395	64.3	756x10⁸	154.0	1810x10⁸	64.4	757x10⁸	154.2	1815x10⁸
S. W. Wall....	0.870	64.0	754	55.6	655	65.4	761	56.8	662
Ceiling.....	3.274	64.8	761	212.2	2491	63.2	747	207.0	2446
Floor.....	2.759	67.8	775	187.0	2138	67.0	771	184.8	2126
N. E. Window	0.194	53.6	695	10.4	135	46.2	655	9.0	127
N. E. Rad....	0.062	124.0	2189	7.7	136	67.0	771	4.2	48
N. E. Wall....	0.614	62.9	747	38.6	459	64.1	754	39.0	463
N. W. Window	0.302	51.3	683	15.5	206	45.8	654	13.8	197
N. W. Rad....	0.078	125.5	2207	9.8	172	66.8	770	5.2	60
N. W. Wall...	2.015	64.5	756	130.3	1623	63.9	752	129.0	1520
Totals.....	12.563[a]	...	.....	821.1	9725x10⁸	...	.....	803.0	9464x10⁸

$\frac{821.1}{4\pi}$ = 65.36 F Avg surface temp

$\sqrt[4]{\frac{9725 \times 10^8}{4\pi}} - 460 = 67.9$ F—MRT

Observed MRT = 69.3 F

$\frac{803}{4\pi}$ = 63.92 F Avg surface temp

$\sqrt[4]{\frac{9464 \times 10^8}{4\pi}} - 460 = 64.0$ F—MRT

Observed MRT = 67.5 F

[a] $12.563 = 3.99\pi$ Radians

over the past twenty years should have made this relationship obvious. A careful study of the work of Bedford and his associates in England in relation to the work of this Laboratory makes such a comparison possible, although arrived at indirectly.

Bedford and others have indicated in their published results a relation between mean radiant temperature and dry-bulb temperature for the same feeling of warmth. From his work, therefore, it is possible to determine the necessary elevation or depression in the dry-bulb temperature to compensate for a counter depression or elevation in the mean radiant temperature observed in an enclosure, while from the work at the Laboratory it is readily possible to determine the elevation in the effective temperature corresponding to a given

rise in the dry-bulb temperature. Combining these two relationships, Table 5 is presented, giving the elevations or depressions in effective temperatures to compensate for given depressions or elevations in mean radiant temperatures of 1, 3, and 6 F. These comparisons are given for a range of from 66 to 85 deg ET, and for three relative humidities, thus including the most important variations in atmospheric conditions found in heating and cooling practice. While these comparisons are given for the necessary *increase* in the effective temperature to compensate for the depression in mean radiant temperature, the same comparison to within the limits of accuracy of the original data applies when the effective temperature of the surroundings is *decreased* to compensate for an *increase* in the mean radiant temperature.

By interpolation it is found that for a condition of 70 deg ET and 30 per cent relative humidity an elevation of 1 deg ET will about compensate for a 2 deg lowering of the mean radiant temperature. For the same effective temperature and 20 per cent relative humidity the same change in mean radiant

TABLE 5—RELATIONS BETWEEN POSITIVE OR NEGATIVE CHANGES IN MAIN RADIANT TEMPERATURE AND NECESSARY CHANGE IN DRY-BULB OR EFFECTIVE TEMPERATURE TO MAINTAIN THE SAME FEELING OF WARMTH [a]

CONDITIONS	E.T.		66	66	66	73	73	73	80	80	80	85	85	85
	R.H.		20	50	80	20	50	80	20	50	80	20	50	80
	D.B.		73	70	67.5	83.5	79	75	93.5	88	83	105	95	86.5
CHANGES	ΔMRT	ΔDB	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET	ΔET
	-1	.98	.63	.66	.68	.53	.56	.60	.43	.47	.51	.37	.41	.46
	-3	2.7	1.75	1.91	1.92	1.46	1.55	1.65	1.19	1.30	1.43	1.05	1.18	1.27
	-6	5.2	3.36	3.48	3.67	2.80	3.00	3.19	2.29	2.50	2.72	1.97	2.19	2.44

[a] Relation between mean radiant temperature and dry-bulb temperature from Bedford's work. Effective temperature relationship based on laboratory's work.

temperature requires a little smaller change in the effective temperature, while with 80 per cent relative humidity a greater change in the effective temperature is required for compensation. Although there is some variation in the compensations in effective temperature required for a given change in mean radiant temperature, it will be observed from the table that these variations are not great. In the column ΔDB the variation in the dry-bulb temperature as taken from Bedford's work is shown. For example, a depression of mean radiant temperature of 1 deg is compensated for by an elevation in the dry-bulb temperature of 0.98 F. In this study it is evident that both the relationship and the measurement of the mean radiant temperature should be further studied as factors in a person's sensation of comfort.

These relationships are somewhat more readily shown in Fig. 13, where the relation between the positive or negative change in mean radiant temperature, and the necessary negative or positive change in effective temperature to result in no change in the feeling of warmth is plotted over a wide range of effective temperatures and for three different relative humidities. These curves are based upon Table 5.

Applying the information from Table 5 and Fig. 13 to the data plotted in Fig. 11, the solid line represents the relationship between the effective tempera-

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ture for comfort and the dry-bulb, mean radiant temperature difference which the correlated work of Bedford and this Laboratory would predict. The slope of this curve is obtained from interpolation on Fig. 13 and the curve itself is so placed in Fig. 11 that it passes through the point 69.0 deg ET and 2.9 F which is the algebraic mean of all the plotted points. Its agreement with the data is fair although the curve as drawn from the points fits the data better.

From Fig. 5 it will be observed that 2.3 deg ET elevation was required in the test in the radiator heated room on January 19 to result in an average

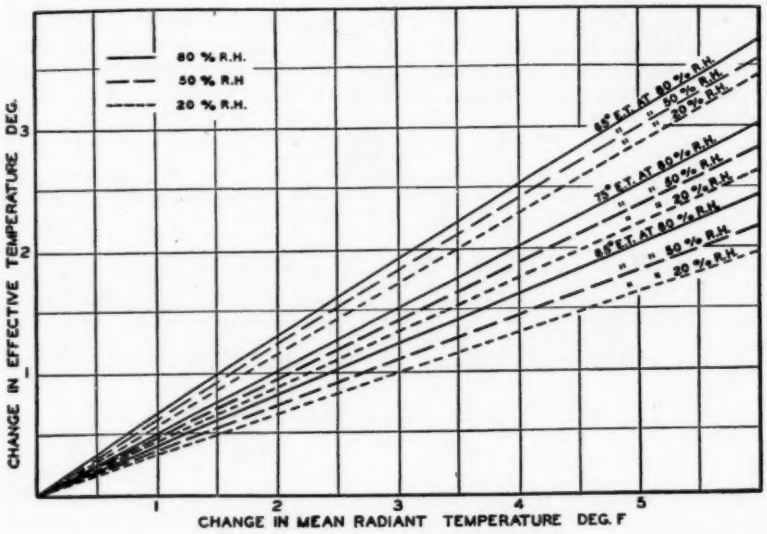

FIG. 13. RELATION BETWEEN POSITIVE OR NEGATIVE CHANGE IN MEAN RADIANT TEMPERATURE AND THE NECESSARY NEGATIVE OR POSITIVE CHANGE IN EFFECTIVE TEMPERATURE RESULTING IN NO CHANGE IN FEELING OF WARMTH

change in the subjects' index of comfort from *4* or *ideally comfortable*, to either *3* or *5*, *comfortably cool* or *comfortably warm*, respectively. Columns L_r and L_c, Table 2, indicate that for the different tests the necessary change in effective temperature to result in a change in comfort index from *4* to *3* or *5* ranges from 1.9 to 3.7 deg ET for the radiation room, and from 1.7 to 4.1 deg ET for the convection room; or an average of 2.7 deg ET change for either room. The first Laboratory study [3] made in 1923 indicated an optimum comfort at 65 deg ET and a required change of 1 deg ET for a comfort change from *3* to *4*. Later Laboratory studies [4] show necessary changes varying from 2.3 to 4.3 deg ET for a comfort change from *3* to *4*, agreeing well with the

[3] See Bibliography 4.
[4] See Bibliography 5, 6, and 8.

results of this study. This relationship between the necessary change in effective temperature for a change in unit index of comfort is closely tied up with the impression which is conveyed to the subjects regarding what they are to interpret as, *comfortably cool,* or *comfortably warm.* Throughout the work of the Laboratory an attempt has been made to have the subjects interpret *comfortably warm* (or *comfortably cool*) as a condition in which they are essentially comfortable but one in which, if a change were required, would be made in the direction of *ideally comfortable.* It is probable that a finer interpretation of this difference was made by the subjects in the earlier Laboratory studies than has been made since.

Considering a change of 3 deg ET as necessary to account for a change of one unit in the comfort scale, it will be observed from Table 5 and Fig. 13 that if no other factors in the environment than the mean radiant temperature were changed the required elevation in mean radiant temperature to result in a change from *ideally comfortable 4* to *5* would be about 6 deg with some variation depending on the temperature and the humidity of the condition. This fact, together with the relatively small difference between the average dry-bulb temperature and the mean radiant temperature in either room, and more particularly in the radiation room, shows clearly that the effect which it was attempted to evaluate in this study was little more than above the threshold of observation and accounts amply for the fact that while this study indicates definite trends when the average results are plotted the small effects could not be very accurately evaluated. Since in this study the mean radiant temperature effects which it was attempted to evaluate were exaggerated over and above what they would be in normal enclosures, *first,* by building these rooms of relatively poorly insulated walls (having a transmission coefficient, U, of 0.19) and *second,* since these rooms had four almost full directions of exposure, rather than the usual one or two exposures found in most enclosures, it will be seen that the small effects observed were large when compared with what should be expected in normal rooms.

In this connection it is of further interest to note that the effect of the radiation was always such as to reduce the difference between the air temperature and the mean radiant temperature. In no case in the rooms in which this study was made did the mean radiant temperature exceed the air temperature, even in tests 18 and 19 when the windows were entirely blanked off in the radiation room with 1 in. structural board insulation so as to reduce to a minimum radiation to cold windows.

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DISCUSSION

G. P. NACHMAN (WRITTEN): The results obtained, in my opinion, certainly warrant the continuation of these experiments and it is to be hoped that they can be continued. There are so many more things that would be not only interesting to know, but constructive from a thoroughly practical point of view.

For example, the use of a 2-speed fan for Room *C*, thermostatically controlled, is certainly an ideal arrangement. In actual practice, however, if such a 2-speed fan were used, some difficulty might be encountered in heating the house. If the duct work were designed for the larger volume of air, then the rooms farthest away from the furnace would receive very little heat with the fan operating on low speed and, of course, if the duct work were designed for the lower volume of air, considerable noise or discomfort might be caused when operating at a high speed in those rooms near the furnace.

Modern practice in residence warm air design usually is based on a register temperature of from 120 to 140 F. The volume of air is therefore reduced, which correspondingly reduces the duct sizes and the cost thereof. The test room apparently had a great many more air re-circulations per hour than is found in the average residence. We know from experience that the lower the temperature of the heating

medium, the less the cost of operation. It is for this reason that hot water heating with a wide range of heating medium is more economical to operate than steam.

In actual practice, the temperature controlling devices in ordinary residential work are generally intermittent in operation. The heat retaining ability of several types of radiators then has an opportunity to make itself felt. One is certainly astounded by the high effective temperatures that had to be carried for *ideal comfort*. Perhaps the authors will be able to give some further explanation of this result, to which they have especially called attention.

One notices that the average air velocity in both of the rooms was unusually high. In Room *R*, between 25 and 30 fpm is higher than one would expect with radiator heat. In Room *C*, from 30 to 40 fpm is also higher than one would normally find in actual residence practice.

Perhaps this may account for the high effective temperatures which had to be carried. If the authors have any explanation of these high air velocities, I am sure such would be extremely interesting.

C.-E. A. WINSLOW (WRITTEN): The investigations conducted by Mr. Houghten and his associates constitute a valuable pioneer contribution to the problem of the respective merits of convection heating and heating by the use of free-standing radiators. It is, of course, only in small part a study of *radiation* since most of the heat given off by a free-standing radiator is in the form of convected heat.

The relatively slight difference between the results recorded in the two test rooms is not unexpected but it is helpful to have the evidence in concrete experimental form. It is natural and inevitable, however, that—as indicated in Fig. 13, a change in mean radiant temperature of 2 deg should produce a change of about 1 deg ET, since air and mean radiant temperatures exert approximately equal influence on comfort.

What is somewhat surprising is that the floor-ceiling temperature and the air movement in the convection room were so close to those obtained in the radiation room. The results, in this respect, demonstrate an unusually successful operation of a convection system in a room of such small size.

Mr. Houghten's comments on the instruments used in measuring mean radiant temperature may, perhaps, give a somewhat misleading impression to those not familiar with this subject. When he says that "100 per cent differences in resultant observations were had at times between the standard Vernon globe and the Laboratory instrument," he must refer, not to mean radiant temperatures but to differentials between mean radiant temperature and air temperature. Since these differentials are only 2 deg or 3 deg, a *100 per cent* variation may mean merely a variation between 1 deg and 2 deg. Whether these small differentials are more accurately recorded by the standard Globe Thermometer or by the Laboratory modification of that instrument, it is not yet certain. The Technical Advisory Committee on Instruments of the Society is making extensive comparative tests on various modifications of the Globe Thermometer in Boston, New Haven, and Urbana. These tests are proceeding harmoniously and satisfactorily, and in a few months we should have a technique for measuring mean radiant temperature agreed upon by all the experts in this field, which can be adopted as a standard procedure.

A. P. GAGGE[5] (WRITTEN): There are two parts to a comprehensive definition of mean radiant temperature: (1) The mean radiant temperature of an environment is the temperature of a uniform enclosure with which an object or person would exchange the same amount of energy by radiation as in the actual environment. (2) Mean radiant temperature refers only to the particular person or object receiving the radiation and therefore is a function of the size, shape and position in the room of that person or object.

[5] Presented for the Technical Advisory Committee on Instruments. D. W. Nelson, *Chairman*; F. R. Bichowsky, L. M. K. Boelter, R. S. Dill, M. K. Fahnestock, A. P. Gagge, J. A. Goff, F. W. Reichelderfer, G. L. Tuve, C. P. Yaglou.

The first part defines the measurement itself while the second qualifies the interpretation of this measurement. With the same wall heated, *e.g.*, no two people are necessarily exposed to the same radiant temperature nor does a man, an eupatheoscope, or a globe thermometer *see* numerically the same radiant temperature. Because of these various possibilities and in order that a common interpretation be placed on all radiant temperature measurements the Committee recommends that the use of the Vernon Globe Thermometer be carefully standardized for this purpose. The Globe Thermometer being symmetrical summates impartially radiation from all directions. Its construction is simple and finally it is portable and satisfactory for field work.

There are five factors one must consider when standardizing the globe thermometer, and in order of importance they are:

(1) The degree of blackening of the globe surface.
(2) The method of measurement of air movement.
(3) The size and weight of the globe.
(4) The measurement of the globe temperature itself.
(5) The measurement of the ambient air temperature.

At present to enumerate briefly, considerable progress has been made in standardizing all these factors except the measurement of air movement. Work along these lines is now under way and in the near future a formal procedure for the measurement of radiant wall temperature with the Globe Thermometer will be presented to the Society.

F. R. ELLIS (WRITTEN): It has been known that convection heated spaces require a higher dry-bulb temperature than radiator heated spaces, the reason for which was generally assumed to be the result of greater air motion rather than the absence of radiant heat.

It was surprising to note how little difference in average air velocity there was in the two rooms. Yet the air velocity in the convection room was frequently 50 to 100 per cent more than that in the radiator room; also, there were recorded 60 per cent more drafts in the convection heated room than in the radiator room for the same number of test periods. Is it not probable that air motion is as much a factor as the mean radiant temperature?

One might expect that the radiator heated room at a lower temperature would require less Btu input than the convection room. The average shown in Table 2 is the reverse of this. However, there is no consistency in the recordings either at the same outside temperature high or low, or for wind velocity or solar radiation. These variations may be found in Table A.

A revised effective temperature chart for spaces with exposed walls and glass for radiator and convection heating would be very much worthwhile.

TABLE A—SELECTED VALUES OF HEAT INPUT

TEST NO.	OUTSIDE TEMP. F	SOLAR RAD.	BTU INPUT ROOM *R*	BTU INPUT ROOM *C*	BTU INPUT *R* IN PER CENT *C*
18	17.2		2341	2973	79
25	17.0		6094	5919	103
15	39.9		1746	1507	116
28	40.2		1666	2161	77
19	13	400	3915	5026	78
25	17	457	6094	5919	103

A. F. NASS (WRITTEN): This paper is interesting, inasmuch as it proves to me that the thought I have had in this matter is true. We have found that in checking over heating and ventilating plants of school buildings, where we had an air circulating heating and ventilating installation; to be comfortable, the temperature reading in the room has always been much higher than in a room where we had radiators installed to take care of the exposed walls, floor, ceiling and glass, and used the air circulation for ventilating.

During the past weeks we have had a great deal of sickness necessitating the closing of schools, etc. It occurred to me that we should go further with this study, and check the various systems installed in our school buildings to determine which system would be the best from a health standpoint.

At present in the state of Pennsylvania they permit the recirculating of 50 per cent of the air. I do not know whether this is correct from a health standpoint.

We have some systems installed in school buildings, using a unit ventilator in a classroom recirculating 50 per cent of the air and discharging the foul air into the corridor, from which it is vented to the outside. It is possible that the venting of the corridor may not be functioning, which would contaminate the air in the corridor which is used by everyone in the school.

The reference to current demands for higher effective temperatures than now shown in THE GUIDE, coincide with information which I obtained at the Construction Industry Conference of the Chamber of Commerce of the United States at Washington, D. C. It was brought out in one of the statements made at this meeting that the boys going to camp at present are not as hardy as the boys that were in the camps during the last war.

W. P. SCOTT, JR. (WRITTEN): I have read this paper with the utmost interest and the most important finding appears to be that in order to obtain a condition of equal comfort, heating by radiation requires a 10 per cent greater heat input than convection heating. This result is especially unusual when it is considered that the effective temperature for radiator heating was approximately one degree less than that for the convection method.

The test, however, did not approximate conditions normally encountered when the heated rooms have only one or two exposed walls. Under such conditions I presume that there would be a lesser difference between the power inputs, and a greater difference in the effective temperatures.

Apparently, this research project is but a start on the problem of radiant heating as compared to convection heating. It would be interesting to learn the results of a comparative test in which the subjects received radiant heat from radiators at all four walls, and one in which panels were the heating mediums. The results of such tests would have a profound effect on present methods of heating. Until such data are obtainable, the conclusion must be that, from an economic standpoint, heating by convection is more efficient than radiator heating.

PRESIDENT FLEISHER: I want to make one criticism which I have of the work of the Laboratory with which, as you know, I have been connected for some indefinite number of years. The Laboratory is much too prone to draw curves through isolated points, seemingly of no relationship to the points at all.

I must say this is the first time I have seen the charts for any of this work. But, certainly, in Fig. 9, I would think from the points that your curve had as much right to recede upwards as downwards; down is the tendency. Personally, I do not think it is necessary to direct our eye any way. We do not want to direct it simply by drawing lines, and if we have no judgment ourselves, why let the Laboratory direct us. For years I have been criticising the Laboratory for drawing too many lines any way they like, through points which have no relationship to the lines.

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R. M. WATT, JR.: I do not want to introduce a discordant note into this development of the Vernon Globe Thermometer. I notice Dr. Gagge said it was well adapted for field use. On the advice of Mr. Houghten we took several of these globes on a field trip on a ship sailing from Key West, and we have taken them on several other expeditions. Our experience does not confirm the statement that they are very suitable for field use. Almost every trip we have had a good many of them dented and broken. We put them up in engine rooms and other heated spaces and the black paint came off. We would be very happy in the Navy if you would develop something which is a little more rugged for measuring this radiant effect.

No. 1173

OPERATING RESULTS OF A RESIDENCE RADIANT WALL HEATING SYSTEM

By E. J. Rodee,* New Haven, Conn.

Introduction

PANEL heating, or low temperature radiant heating, has been popular in continental Europe and the British Isles for some years, but only recently has such a method of heating received much consideration in this country. The majority of installations which have been made use either the floor or ceiling as the radiant surface, very rarely the side walls. However, it would seem that to assure maximum comfort to the occupants, heating the side walls might be more desirable. When the body is in a standing or sitting position a larger percentage of the area of the body is radiating directly to vertical surfaces such as the side walls rather than to the horizontal floors and ceilings. Clarence M. Woolley has, for many years, been fathering an interest in radiant heating. It was mainly because of his suggestion that a wall heating system was devised and a study made of its operation in a residence.

Description of Wall Heating System

This wall heating system was designed to heat the air in the stud space of either outside or inside walls, the air, in turn, heating the plaster which becomes the radiation surface. Cast-iron convector radiators were designed to fit between the studs and so arranged that by removing the base board the heating sections could be removed without disturbing the plaster.

These radiators were of two types, a double type section and a single type section. The double type was "L" shaped for use in inside walls while the single type was a straight vertical section with fins on one side only for installation in either outside or inside walls. These sections were 13 in. long and connected by 1-in. pipe nipples. Different length pipe nipples were used in connecting these sections to allow for variation in stud spacing. The single type section weighed 7½ lb and its heat emission rate was 2 sq ft of equivalent

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Presented at the 47th Annual Meeting of the American Society of Heating and Ventilating Engineers, Kansas City, Mo., January, 1941.

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direct radiation, while the double type section weighed 14 lb and its heat output was 3½ sq ft of steam. In comparing the weights of these radiators with tubular type radiators, they were about 25 per cent less for the same heating effect.

A baffle was installed within the stud space behind and above the radiator to separate the upward and downward currents of air, and cause the air to circulate. Fig. 1 shows a section of a wall with the heating element and baffle

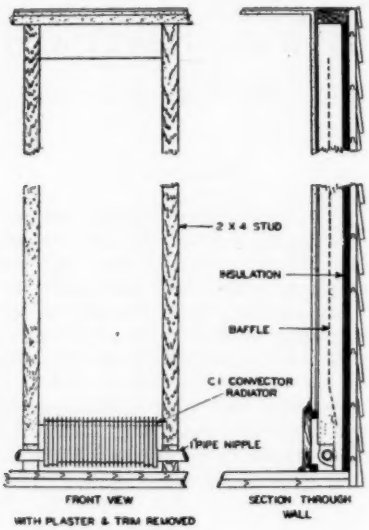

FIG. 1. DIAGRAM OF WALL HEATING INSTALLATION

in place. Fig. 2 shows a front view of a wall before the lath and plaster were applied. Fig. 3 shows convector radiator installed.

Prior to this installation, experiments had been made to determine the best type of lath for the heated walls. It was found that either rock lath or metal lath gave equally good results. The plaster used was a standard mix of lime and sand, and no special precautions were taken in its application. All the walls were papered after completion. The unusual feature of this installation was that steam instead of hot water was the heating medium. This was possible because the heating elements were not in direct contact with the plastered walls. The warm air within the stud space heated the walls uniformly and thereby caused no cracking of the plaster, as is sometimes the case where plaster is applied directly to the heating surface when pipe coils are used to convey the heating medium.

This heating system was installed in all the rooms of the first floor (except the kitchen) of a two-story frame house located in West Haven, Conn. The remainder of the rooms, including all the second floor, were heated with free standing radiators. Fig. 4 shows the exterior view of the house, and Fig. 5 is a first floor plan showing the walls which were heated. The building is a two-story structure with a basement and attic. The calculated maximum design heat loss of the building was 74,760 Btu per hour, using an outside temperature of −5 F, and an inside temperature of 70 F. The design heat loss by conduction was 57,200 Btu per hour and 17,560 Btu from air infiltration.

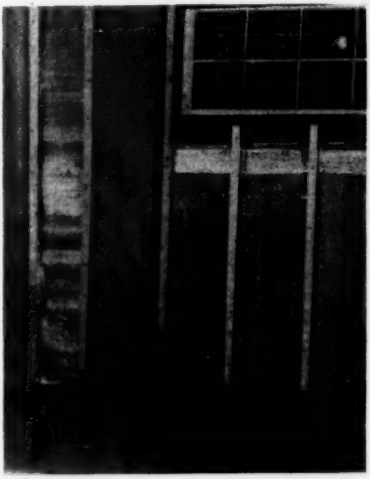

FIG. 2. PICTURE OF WALL SHOWING CONVECTOR RADIATORS AND BAFFLE IN PLACE BEFORE LATH AND PLASTER WAS APPLIED

The second floor ceiling was insulated with 4-in. mineral wool between the joists and there was no floor in the attic, giving a coefficient of 0.08 Btu per square foot per hour per degree Fahrenheit temperature difference. The side walls, except those used for heating, were of the usual frame construction, uninsulated, with a coefficient of heat transmission of 0.26. The heated side walls were insulated with 3 layers of aluminum foil mounted on corrugated asbestos paper attached to the sheathing facing the stud space. This gave a coefficient of 0.16 between the air in the stud space and the outside and 0.125 between the room air and the outside. More insulation would have been desirable, but this was the maximum amount which could be installed in a 4-in. stud space and still leave room for proper air circulation within the space itself.

The wall heating elements including the free standing radiators were hooked up as a one-pipe steam system supplied with steam by an oil-fired steam boiler.

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The system was controlled by a heat-anticipating type thermostat mounted on an inside wall of the living room.

TEST METHOD AND EQUIPMENT

In order to make a direct comparison between wall heating and free standing radiators, the rooms which were equipped with wall heating were also equipped with radiators. These radiators were 3-tube, 23 in. high and 2½ in. from center to center of sections. The radiator heating system was used on alternate weeks during the test period. All the radiators and the wall heating sections were equipped with tight closing valves so that when one system was in use, the other could be shut off.

The oil supply to the burner was so arranged that daily oil weights could be taken. These weights were taken near midnight so that any sun effect on

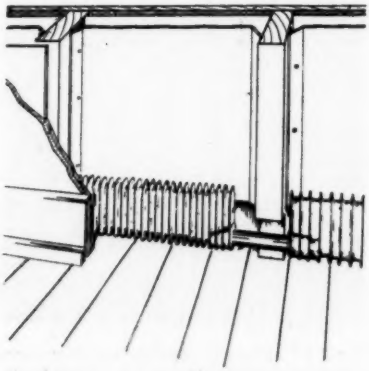

FIG. 3. CLOSE-UP VIEW OF CONVECTOR RADIATOR

the structure would have been dissipated. Duration and number of cycles of burner operation were recorded. Inside temperature and humidity were automatically recorded in the various rooms in the house.

The outside weather conditions were recorded by a combination temperature and humidity recorder located about 20 ft from the north side of the house and 6 ft above the ground. Wind movement was recorded by a three-cup anemometer 6 ft above the peak of the roof and connected electrically to a counter in the basement. The number of hours of sunshine were taken from records of the U. S. Weather Bureau Station, New Haven, located about 5 miles north of the house under tests. Inside wall surface temperatures were measured by a thermopile, and air temperature gradients were measured by iron-constantan thermocouples, using a portable potentiometer accurate to ± 0.2 F.

FIG. 4. HOUSE IN WEST HAVEN, CONN., WHERE WALL HEATING STUDIES WERE CONDUCTED

Periodic checks were made on the boiler burner unit to make certain that the efficiency of the unit did not change. These measurements consisted of flue gas analysis, flue gas temperature, and draft measurements. The flues of the boiler were inspected regularly and cleaned when necessary.

TEST RESULTS

The two curves for the two methods of heating, giving the daily oil consumption plotted against indoor-outdoor temperature differences, are shown in Fig. 6. It will be noted that the fuel oil consumption is approximately the same for the two methods of heating. Fig. 7 shows the fuel consumption per 24 hours per degree temperature difference between inside and outside, plotted

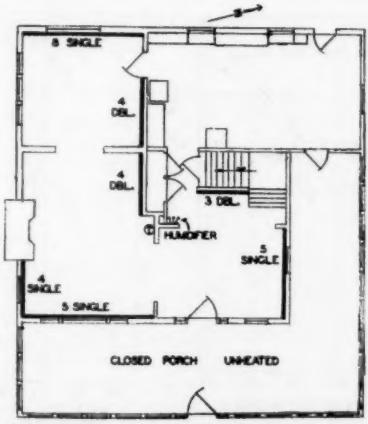

FIG. 5. FIRST FLOOR PLAN SHOWING HEATED WALLS AND NUMBER AND TYPE OF CONVECTOR RADIATOR SECTIONS IN EACH WALL

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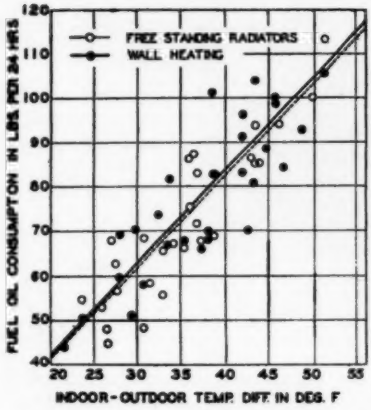

FIG. 6. FUEL OIL CONSUMPTION FOR WALL HEATING SYSTEM AND FREE STANDING RADIATORS WITHOUT CORRECTION FOR DIFFERENCE IN OUTSIDE WIND MOVEMENT

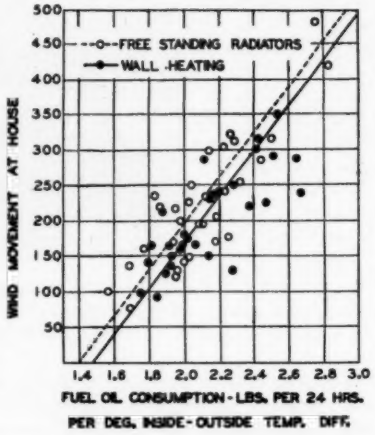
FREE STANDING RADIATORS
WALL HEATING
WIND MOVEMENT AT HOUSE
FUEL OIL CONSUMPTION - LBS. PER 24 HRS. PER DEG. INSIDE - OUTSIDE TEMP. DIFF.

FIG. 7. EFFECT OF WIND MOVEMENT ON FUEL CONSUMPTION

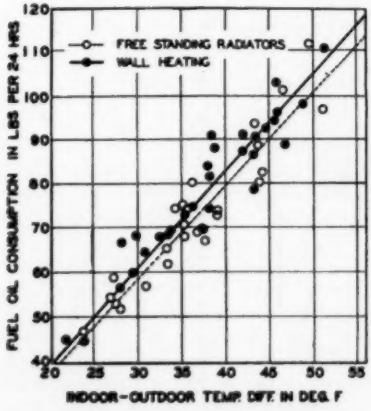

FIG. 8. FUEL OIL CONSUMPTION FOR WALL HEATING AND FREE STANDING RADIATORS AFTER CORRECTING ALL POINTS TO 200 MILE WIND PER 24 HOURS

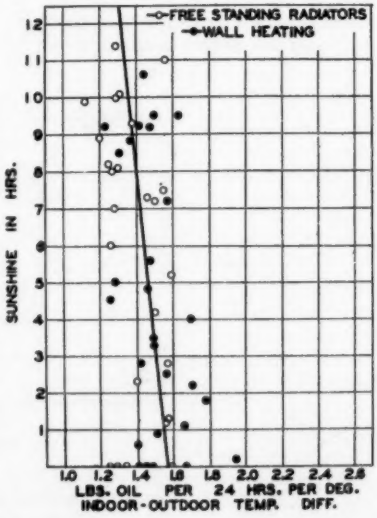

FIG. 9. EFFECT OF SUNSHINE AS RECORDED BY THE U. S. WEATHER BUREAU ON FUEL OIL CONSUMPTION

against miles of wind movement per 24 hours. Both curves have the same slope and the fuel oil consumption is increased 0.3 lb per degree difference in temperature per 24 hours for an increase of 100 miles of wind movement per 24 hours. The curves in Fig. 8 are the same as shown in Fig. 6 except that all the points are corrected to 200 miles wind movement (approximate average wind movement of all tests) per 24 hours. This shows that the wall heating system required about 4 per cent more fuel than the system using free standing radiators, when corrected for outdoor wind velocity. The reason this difference appears in these curves and not in Fig. 6 is because the average wind movement during the tests with radiator heating was greater than during the wall heating tests.

An attempt was made to determine the effect of sunshine on fuel consumption. This is shown by Fig. 9, where hours of sunshine are plotted against

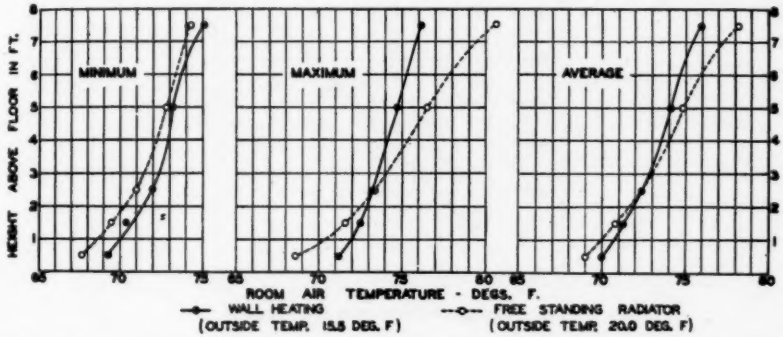

FIG. 10. COMPARISON OF AIR TEMPERATURE GRADIENTS DURING DIFFERENT PHASES OF A HEATING CYCLE BETWEEN WALL HEATING AND FREE STANDING RADIATORS

pounds of oil per degree difference in temperature, corrected to zero wind movement. While there is a definite trend of the points indicating that a slight saving (about 0.02 lb of oil per hour of sunshine per degree difference in temperature) is effected, the amount is so small and the spread of the points so great that this relation cannot be considered significant when compared with the wind effect.

The difference between the floor-ceiling temperature gradients is shown by the curves in Fig. 10. The first set shows the minimum temperatures in the middle of the living room, just before the air temperature begins to rise. These temperatures occurred after the thermostat called for heat and at the instant steam came to the radiator inlet. The second set shows the maximum temperature occurring in the heating cycle taken at the same location. The last set of curves shows the average air temperatures at the different levels during several heating cycles. These temperature readings were taken every two minutes, then averaged so that they gave an average tmperature in respect to time and not an average between maximum and minimum temperatures. The ther-

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mostat setting remained the same for all tests with the two different methods of heating. However, in order to maintain the same feeling of comfort it was found necessary to maintain a 2 F higher air temperature with the radiator heating as compared with the wall heating. Fig. 11 shows how lowering the inside air temperature 2 F with the wall heating method would affect the fuel oil consumption when compared with the fuel oil required for heating with free standing radiators under the same outdoor weather conditions. While to be correct these curves should have been plotted with fuel oil consumption against outdoor temperature, they were plotted using the same difference between indoor and outdoor temperatures as in Fig. 8, so that these two sets

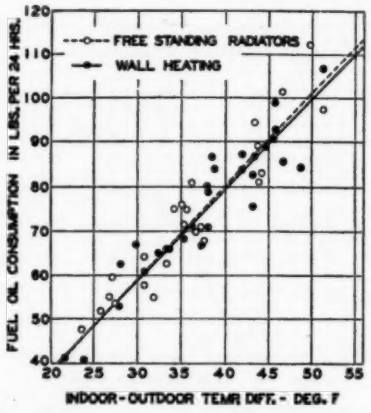

FIG. 11. FUEL OIL CONSUMPTION FOR WALL HEATING AND HEATING WITH FREE STANDING RADIATORS AFTER CORRECTING TO THE SAME COMFORT CONDITIONS

of curves could be compared. The difference in the air temperatures at the lower levels, as shown by the curves in Fig. 10, may easily account for part of the difference in the feeling of comfort.

Fig. 12 is a group of curves showing wall surface temperatures taken at different levels from floor to ceiling. These curves show the changes in temperature of the wall surfaces during a heating cycle.

SUMMARY

Wind movement appreciably affected the heat loss or the fuel consumption of the house in which the tests were conducted, and would lead one to believe that, in conducting tests in which the heat loss of a structure is the controlling factor, wind velocity readings are necessary.

The saving of fuel as affected by sunshine on this house was not significant if the hours of sunshine, as recorded by the Weather Bureau, are taken as an index of the heating effect of the sun; but this factor, of course, ignores the intensity of sunshine.

A wall heating system can be installed in the outside walls of a house without increasing the wall thickness and without adding more than a normal amount of insulation, and still compare favorably with free standing radiators from a standpoint of cost of operation.

While no appreciable saving in fuel was noted by this study, it must be borne in mind that all the wall heated rooms were on the first floor beneath a heated

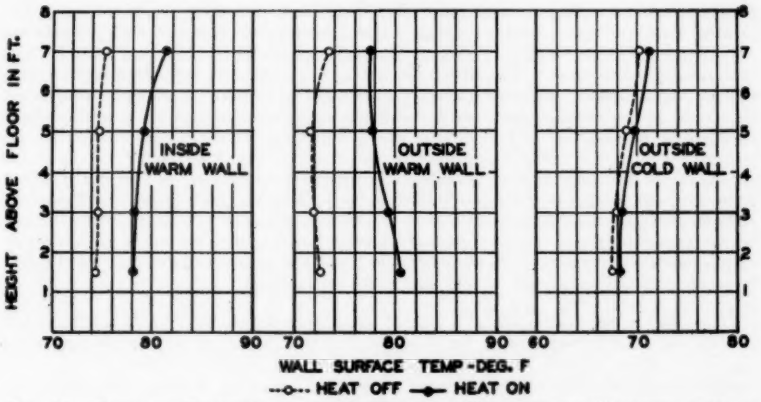

FIG. 12. SURFACE TEMPERATURES OF DIFFERENT WALLS WHEN HEATING WITH A RADIANT WALL HEATING METHOD

space. If an unheated attic had been above these rooms, the lower air temperature gradient obtained with the wall heating system might have effected a fuel saving.

The temperature diffeernce between floor and ceiling is materially less when heat is supplied by the walls than when heating with free standing radiators. The air temperature near the ceiling remains nearly constant with the wall heating system, between 75 and 76 F, regardless of the point on the heating cycle, while with radiator heating the ceiling air temperature is approximately 7 deg higher at the maximum point of the heating cycle than at the minimum point.

DISCUSSION

T. NAPIER ADLAM (WRITTEN): The author offers many interesting details in the subject of radiant heating and his paper is particularly useful because it brings out several salient features in the relation between radiant wall heating and ordinary

cast-iron exposed radiator heating. However, I would like to explain that applying radiant heating to the wall is not as rare as is suggested. When radiant heating received its inception in the British Isles, the walls were often chosen as the place to install the imbedded pipes. Even the Romans as far back as 50 B.C. used the walls as well as the floor as heating surfaces and formed ducts in same through which they conveyed the hot gases from the charcoal fires and thereby obtained the necessary heat for their comfort.

Generally speaking, it appears that the only draw back to the heated wall method, is the possibility that walls thus treated may often be shielded by furniture or other fittings, whether it be an office building or a residence. When there is nothing on the surface of the wall to indicate where the heated surfaces are located, it often happens that furniture and fittings are moved around in the summer time. Then when cold weather arrives, the rooms are found to be uncomfortably cold. I find this happens in our own factory office where we have a number of heated panels situated in places which now offer ideal spots for filing cabinets, etc. The relative merits of floor, wall, or ceiling heating have never been definitely established beyond the fact that certain rooms or buildings will offer greater facilities for one type more than another. Then again, those heating engineers who first started with the idea of imbedding pipes in plaster have generally adopted the method of placing the pipes in the ceiling; whereas those firms using metal panels invariably adhere to the wall heating.

One of the principal reasons for using the ceiling as a heating surface is that there is no fear of the surface being shielded by any obstacle which would intercept the radiant rays leaving the panel. When the radiant surfaces consist of metal panels, which may either be in the form of cast-iron flat plates with water ways or fabricated metal sheets with pipe coils spot welded to the back of plate, these can be attached with no difficulty whatever to the side walls of a room.

It is safe to say there are (or shall I say, were before the bombing) hundreds of schools, churches and other buildings in England heated with radiant panels on the side walls. Where large windows exist, it is a distinct advantage to have a radiant panel installed on the side wall under the windows, because this takes care of the extra cold surface presented by the windows and prevents any down draft. Floor heating is often recommended as being cheaper in construction. It is also recommended in buildings where there are difficulties in placing the full amount of heated surface on the side walls and where the rooms are too high for ceiling heating to be effective.

In the British Isles the floor heating method has been exclusively adopted in those schools where instead of the outside walls being constructed in brickwork or concrete, they have moving partitions which can be opened to give plenty of fresh air. I think it is correct to say that all three methods have received an equal share of attention in developing this system of warming, but naturally, there are certain heating engineers, architects, and contracting firms who will favor one or the other according to their own particular fancy.

The particular method illustrated in this paper will, I think, prove somewhat inefficient because of the close proximity of the warm air space to the outside surface. Many systems have already been installed with warm air passing through ducts formed in the side walls as well as in the floor; but from my experience these have shown a greater loss of heat due to the transmission of heat through the walls. I think this is the real reason why the fuel consumption does not show up better for the heated wall system. In Europe where wall heating is now generally dealt with by iron panels, the heat losses through the wall are reduced to a minimum by using very thick pads of pressed granulated cork or asbestos mattresses. It is very

rare that warm air is used in the walls although I have designed a few such systems for operating amphitheaters in connection with hospitals. In these instances specially constructed walls were provided to reduce the loss of heat to a minimum.

The charts presented are also interesting, although I am afraid it is unsafe to base any definite opinion on the results representing the fuel oil consumption in Fig. 6 for the reason stated. Then again, when dealing with radiant heating systems, comparative results of fuel consumption based on temperature differential between inside and outside only lead to an erroneous conclusion. If a radiant heating system is designed, installed and operated correctly, it should not be necessary to have the same air temperature as with cast-iron radiators, because the mean radiant temperature in the former will be higher. This makes possible a lower air temperature. Therefore, to get a true relation in fuel consumption, the comparison should be considered on a question of comfort conditions rather than temperature differential.

If we refer to the chart in Fig. 7, which indicates the relation of fuel consumption to wind velocity, it seems difficult to form any true conclusion on this point because there is such a small portion of the radiant heat panels on the exposed outside walls. Even after making a correction for a constant wind velocity, there stilll remains some ambiguity about the results. It will be noted in Fig. 5 that most of the radiant heat surface is either placed on inside walls or on walls protected by a closed porch. Even though the closed porch is not heated, it does protect the walls from both sun and wind effect. This is a good practice and is recommendable for such a building, but when comparing test results I do not think it wise or even safe to draw such a general conclusion which might be entirely different in another building without a closed porch. It may have been more helpful if the inter-change of air in the rooms had been known, because although the wind velocity would affect the heat losses, the infiltration depends on its direction as well as its velocity. This, of course, would be very difficult to determine, but without this information I think it would be unwise to form an opinion as to the relative effect of wind on the two systems. It is only natural to expect, however, that the effect of the wind on the outside *heated* walls will be more serious than on the outside *unheated* walls.

Although there is a layer of insulation material between the warm air space and the outside weather boards, it will be apparent from Fig. 1, that a much greater temperature differential exists where the walls are heated. It is not given by the author, but I think it can be safely assumed that the warm air in the wall space will not be less than 100 to 110 F, whereas with the walls not heated, the air temperature close to the wall will be slightly less than that of the room, not more than 70 or 72 F. Consequently, I do not think it is possible for anyone to make a very accurate deduction from the figures presented.

It seems a little doubtful as to whether the figures given in Fig. 9 represent the true effect of the sun, simply because the closed porch protects most of the heated walls from the effect of the sun and because there is very little heated surface exposed to the sun during the middle of the day.

It has been my experience that a greater radiating effect may take place depending on the upper stratosphere. There are periods during both day and night when radiation may take place from the building to the upper stratosphere, while at other times radiation may take place from the upper stratosphere to the building. A gain of heat by radiation may take place even when the surrounding atmospheric temperature is quite low, and since this can take place during the long hours of the night, it may have a more important bearing on the fuel consumption than the winter sun which may only shine with any degree of heat for a few hours.

The curves shown in Fig. 10 are very interesting, since they show the changes which take place with a system controlled by the *on* and *off* method. They also

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prove very conclusively that a radiant heating system is affected less by changes of heat and gives a more consistent condition of comfort than a system with exposed cast iron radiators. A convected heat system would undoubtedly have shown even greater fluctuations than the free standing cast iron radiators in this system.

It would have been interesting to see the data obtainable if the tests had been made with a steady control, even though the heat had been regulated manually, since it is always difficult to get a true average with the *on* and *off* method when we get big oscillations. Such oscillations are definitely shown in Fig. 10 by the three dotted curves for free standing radiators. It is worthy of note that the temperatures taken at a height of 7 ft 6 in. and indicated by the top points of the three solid line curves in Fig. 10 for radiant heat remain practically constant for the *on, off,* and average heat; whereas for the free standing radiators, the corresponding points vary from 74.2 to 81 F. This, I think, proves again that when the walls or other parts of the room structure are warmed for radiant heating, the temperature will remain comfortably consistent for long periods, even though there are changes in the outside temperature and regardless of the fact that the heat is turned *on* and *off* intermittently at fairly wide intervals.

To cite an extreme case which proves this characteristic, the large Cathedral at Liverpool, England which I designed as a floor heating system some years ago, takes 36 hours for the air temperature to drop 1 deg after the heat has been shut off. This, of course, is an exception and due to the very heavy masonry.

It is worthy of note that in Fig. 10 the air temperature as recorded for radiant heat never dropped as low nor went as high as the air temperatures recorded for cast iron radiators, proving that a more moderate heat is obtainable by this method of heating.

It is stated in this paper that a wall heating system may be installed in the outside walls of a house without increasing the wall thickness and without adding more than a normal amount of insulation, and that the results will still compare favorably with free standing radiators. I think the insulation of a building is very important at all times because the more protection we can place in the walls and roof of a building, the less heat will pass through, and the warmer will be the inner surface, regardless of the type of heating. Since warmer inner surfaces mean a natural higher mean radiant temperature, it means less heated panels necessary to maintain the correct mean radiant temperature and at the same time promotes greater comfort with a lower air temperature. I think if the walls had been heated in a different way, so that less heat passed through to the outside, it would have shown a con siderable saving in fuel cost to the advantage of wall heating. There is nothing wrong with the heated wall method for radiant heating, provided the heated surfaces are not shielded by furniture, or other obstacles, and that the transmission loss is reduced to a minimum.

C. F. BOESTER (WRITTEN): This paper presents a discussion of a method of providing human comfort that is intensely interesting to all of us, for it has been quite sometime since the heating industry as such has made any fundamental changes in the means of providing winter comfort. Because of radiant heating some basic changes in our methods of building construction will be required—changes for the better which will contribute much to lower building costs. Furthermore, such changes will be absoutely essential if we are to have the rock bottom heating costs which are possible with this method. The changes required in buildings, which fortunately are rather simple to accommodate such heating, will also permit and justify the use of this method of heat exchange to provide summer comfort with the same equipment.

To me the outstanding point developed in Mr. Rodee's studies is the tremendous effect wind movement has on heating costs. It surely indicates that researches in

this direction are really warranted. It should be interesting to ask the author what the air velocity was through the heating panel and, of course, whether or not the air volume was determined. What was the air temperature leaving the convectors and what was its temperature upon entering the convectors, and what was the air temperature in the top of the panel? One last question—why would not equal or better results be had by letting air enter at the baseboard and leave by an opening at the top, thereby inducing air motion in this room? This would also permit heavier insulation in the wall.

F. E. GIESECKE (WRITTEN): Two important results were secured. It was found that there was no material difference in the cost of heating by the two methods for equal comfort conditions, and that it was necessary to maintain an air temperature 2 F higher for radiator heating than for wall panel heating in order to provide the same feeling of comfort with the two methods of heating.

Among the advantages claimed for panel heating by its advocates are higher floor temperatures with resultant greater comfort, particularly for small children, and lower air movements with resultant greater purity of air because dust particles, to which pathogenic germs may be attached, will settle out more readily; this is of especial importance in hospitals.

I hope that Mr. Rodee will continue his study and determine, for the two methods of heating, the respective air movements; wall, floor, and ceiling temperatures; and the mean radiant temperatures; also, the influence of mean radiant temperature on feeling of comfort.

G. P. NACHMAN: What do you use for the baffle?

MR. RODEE: Corrugated cardboard faced with paper on both sides. In a second installation 30-gage sheet-iron was used for the baffle. Nearly any type of smooth sheet material can be used which is easy to apply and will hold its shape. There are no forces acting against this baffle and as it is in a heated space there will be no free moisture present to cause deterioration or corrosion.

PRESIDENT FLEISHER: This is a subject which is most interesting, but we are too impulsive and go from one type of thing to another without sufficient consideration. We are rushing into radiant heating with the idea that England has given up all other methods. That is not true at all. In most cases in England, in order to compensate for rapid variations in outside conditions, they have supplementary heat such as fire places, where most of the Englishmen stand and let their guests freeze in the open room. I would like to ask the author whether the same effect that he claims for radiant heating could not be obtained through insulation, so that the walls eventually take on a uniform temperature, and whether there is a possibility of no *sun effect* being due to the fact that insulating against the escape of radiation or radiant energy, might not prevent radiation from coming in by the same means.

MR. RODEE: We can insulate our houses and it will be very effective but at no time can you sufficiently insulate the outside walls and bring those surface temperatures above air temperatures in the room; whereas, with wall heating or low temperature panel heating you do bring those temperatures above air temperatures, and thereby counteract the effect of the cold window surfaces. You can increase the inside temperatures of those window surfaces by addition of storm sash or double or triple glazing.

PRESIDENT FLEISHER: If you insulate against your radiation escaping, would you not insulate against radiation coming through?

MR. RODEE: It is true that an insulated wall will prevent heat entering from the outside caused by the sun as well as preventing heat escaping from the inside to the

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outside. Glass areas, however, act differently because of the different wavelength of the sun radiation and the radiation from the warm surfaces inside the house to the outside. The radiant heat from the sun can pass through the glass to the inside, but the radiant heat from the warm surfaces inside the house cannot directly pass out through the glass because of the wavelength. This house was not oriented particularly well to benefit by receiving heat from the sun through the glass areas in the winter time.

MR. NACHMAN: One of those rooms, I noticed, has the wall panel installation on the inside wall. The practical installation of such a system would be much easier on inside walls. Do your results give any indication whether comfort results could be obtained by working from inside the wall as well as applying the panels to the outside wall?

MR. RODEE: We did not have enough surface. That is one thing we hope to do sometime is to have a complete house where the inside and outside walls are heated, and there are free standing radiators. Then we can definitely determine what the comfort effect would be. My guess is you would not obtain as desirable a comfort condition with the inside walls heated as you would when heating the outside walls.

With the outside walls heated, you can sit in any part of the room and that means right up next to an outside wall. Unless the building is well insulated, if you sit very close to the outside wall you are uncomfortable with the radiator heating system.

J. B. FULLMAN: I just wanted to ask a question of structural strength in that system. I noticed you notched the studs to allow the pipe nipple to pass from radiator to radiator. I imagine the nipple is about 1½ in. in diameter.

MR. RODEE: It is 1¼ in.

MR. FULLMAN: In a bearing wall how would you compensate for that?

MR. RODEE: If a stud in a bearing wall is assumed to be a column, you can theoretically have the stud come to a point on either end, tapering from the center. With the usual spacing of studs on 16-in. centers, I think, you will find you have about three times more strength than is actually needed for the support of the building.

I think stud spacing was based on 16 in. centers because of older methods of applying lath and plaster to prevent cracking. If some of you recall, we had quite a severe hurricane up in that section about two years ago. This system was already installed and is still in operation.

Another wall heating system was installed shortly before this hurricane, where both the first and second floor outside studs, were notched and the hurricane hit that particular house harder than it did where the test house was located, and that house is still in very good condition, the walls remaining perfect.

PRESIDENT FLEISHER: Did you have any days in which you had wide variations of temperature, when you had to do like most of the people who have radiant heating, in England, throw their windows open in order to get the rooms cool? That is, where the walls, being large surfaces, stored up heat at a certain temperature. I found in London, for instance, that with the wide variation of 30 to 70 F, which they get on most clear days during the winter, everybody threw their windows open. How would that effect your totals?

MR. RODEE: We did not have any trouble on that point. You have to consider it is the usual practice, as I understand, in radiant heating or low temperature panel heating systems in England, to heat a large mass of material including the

heavy masonry walls to a comparatively low temperature. This large mass of material must cool during a long period of time. I believe, Mr. Adlam mentioned in his written discussion that he installed a floor heating system in a Cathedral in Edinburgh, where it took 36 hours for the air temperature to drop 1 deg after the heat was turned off.

If the plaster alone is heated, and not the heavy masonry walls, as are often found in public buildings, the heat storted in the plaster is not great. While the weight of the plaster is quite high its specific heat is low so consequently a comparatively small amount of heat can be stored this way. Proof of this is shown by the number of heating cycles which occurred with the two types of heating systems. The radiator heating systems had about 10 per cent more heating cycles than the wall heating system. This shows that the temperature drop inside the house is nearly as rapid when the wall heating system is used as with a radiator system.

I wish to emphasize that this wall heating system used steam as the heating medium and not forced hot water as nearly all low temperature panel heating systems use. This is made possible by the method of installation, the heating elements do not come in contact with the plastered walls, only the warm air in the stud space comes in contact with the plaster. The baffle causes the air to circulate within the stud space, the warm air rises in front of the baffle next to the plaster and the cooler air drops down behind, between the baffle and the outside wall insulation. This system is much more efficient than one where warm air is blown through the stud space because the air is at a much higher temperature next to the plaster than it is near the outside wall insulation, where as when air is forced through the stud space by a fan the air is the same temperature next to the wall insulation as it is next to the plaster.

Referring further to Mr. Adlam's discussion I wish to apologize if my paper gave the impression that radiant wall heating was original with me. I only wished to point out that radiant heating as installed at the present is usually installed in either the ceiling or the floor. When the heat loss from the heated space is high, because of ventilation and glass area, high temperatures must be carried on these heated surfaces to compensate for the heat loss. If these temperatures are high, either on the floor or on a low ceiling, discomfort will be felt by the occupants, whereas if the heated surfaces were vertical side walls, even though the surface temperatures were high, the occupants would still be comfortable.

The placing of furniture tight against the side walls will tend to lower the heat output of the wall area which it shields, but if this furniture is placed an inch away from the heated wall so that air can circulate between it and the heated surface, the heat output from the wall will not be lowered materially.

As to the heat loss to the outside by this warm wall method of heating, Mr. Adlam has apparently overlooked the true function of the baffle. The baffle, by separating the warm air currents in front of the baffle from the cooler air currents behind the baffle next to the cold outside wall, prevents hot air from coming in contact with the cold surfaces. The average air temperature in front of the baffle is 30 to 35 F higher than it is behind the baffle next to the cold wall. In systems where warm air is forced through ducts in a cold wall it is naturally impossible to have the air temperature higher next to the surface which is being heated and the outside surface of the wall from whence the heat escapes. In order to prove the efficiency of this system it would be necessary to have an entire house heated in this manner and the same house compared with a conventional type heating system. We hope to be able to do this in the near future.

Mr. Adlam's discussion on the radiating effect caused by the condition of the upper stratosphere is very interesting. Radiation will occur from a heated structure

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at night regardless of the condition of the upper stratosphere; however, the radiant heat loss will be somewhat greater on clear nights than on cloudy nights. The amount of this radiation loss is, of course, dependent upon the outside surface temperature of the building, which temperature will vary from 1 to 6 F above air temperature when the outside air is 0 F, depending upon wind movement and degree of insulation. During daylight hours the building will receive radiation from the sky even though the sun is not visible and clouds are very heavy.

I agree with Mr. Adlam's statement that the insulation of a building is very important and that in an installation such as described in this paper 2-in. by 6-in. studs would have been justified from an economic standpoint so that additional insulation could have been added.

Mr. Boester has asked some interesting questions and I am sorry that I am unable to answer all of them. No air velocity readings were taken in the space between the baffle and the inside surface of the plaster, but from observation by smoke tests, I would judge it to be from 150 to 200 fpm. The maximum temperature of the air leaving the convector was 140 F and the air temperature behind the convector just before it entered to be heated was about 70 F. The air temperature at the top of the baffle was about 90 F.

If an inlet and outlet were made at the bottom and the top of the panel into the room half of the advantage of the panel heating system would be defeated. The increased air movement in the room would cause more dirt to be distributed in the atmosphere. The air temperature differential between floor and ceiling would be increased thereby nullifying one of the important advantages of panel heating.

No. 1174

COMFORT REQUIREMENTS FOR LOW HUMIDITY AIR CONDITIONING

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Gas Association*

THE evaluation of summer cooling and air conditioning requirements has been under consideration for the past several years by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The studies [1-6] to date have dealt with the determination of these requirements with moderate relative humidities and for persons of varying ages, both sexes, and in different geographical locations. In most cases no particular effort was made to maintain any specified relative humidity, but percentages ranging from 40 to 80, with the maximum number of tests at about 50 per cent, have prevailed.

Recently the Committee on Research developed plans for a study designed to give comfort requirements for persons in what is generally known as the low humidity field. This paper reports the results of the Laboratory's study at 30 per cent relative humidity with some comparison in the higher humidity conditions, and a few exploratory tests in air conditions having humidities as low as 15 per cent. The study was made in the psychrometric rooms of the Laboratory in Pittsburgh under the direction of the Technical Advisory Committee on Sensations of Comfort † and much the same technique as that reported [1, 2, 7] in earlier studies was employed.

In order to make it possible to maintain the desired low humidities during the heat of the summer, a conditioning unit utilizing a dehydration agent was installed in connection with each of the two psychrometric rooms. A diagrammatic sketch showing the relation between one of the two psychrometric rooms,

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[1-6] See Bibliography.

† ASHVE Technical Advisory Committee on Sensations of Comfort: Thomas Chester, *Chairman*, C. R. Bellamy, E. P. Heckel, W. S. Kilpatrick, Dr. W. J. McConnell, A. B. Newton, B. F. Raber, C. Tasker, L. C. Soule.

Presented at the 47th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., January 1941.

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the conditioning equipment, and a number of pertinent arrangements in the room is given in Fig. 1. Return air from the room passed through the dehumidifying unit where its moisture content was reduced and its temperature raised. The hot, dry air then passed over cooling coils (through which water cooled by a 4-ton ammonia refrigerating plant was circulated), then over direct expansion dichlorodifluoromethane coils supplied with refrigeram from a 2-ton compressor, through a blower and finally into the plenum chamber of the psychrometric rooms where it passed over thermostatically-controlled electrical heating coils of sufficient capacity to give a 5-deg temperature rise. In order to maintain the moisture content of the air in the psychrometric rooms

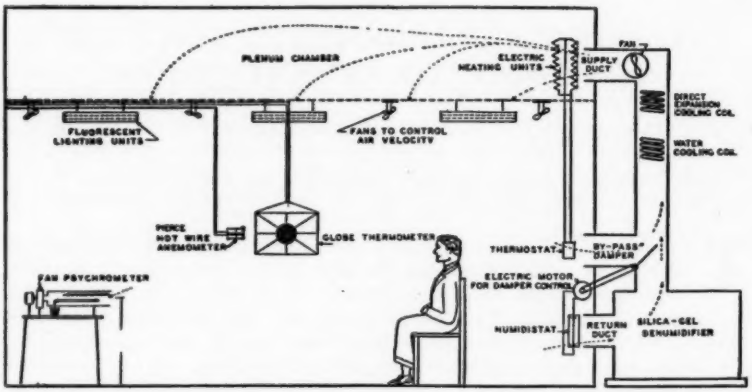

FIG. 1. DIAGRAMMATIC SKETCH OF ONE OF THE TWO PSYCHROMETRIC ROOMS, SHOWING ARRANGEMENT OF CONDITIONING SYSTEM, CONTROLS, LOCATION OF ONE OF SUBJECTS IN THE ROOM, AND OTHER PERTINENT ARRANGEMENTS

at the desired level, a humidistatically-operated by-pass damper controlled the volume of room air passing through the dehumidifier. Thus it was possible to control independently the moisture content and the dry-bulb temperature of the air. With this arrangement, relative humidities of 30 per cent could easily be maintained during the most severe summer conditions with 6 persons in the room. Minimum relative humidities of from 15 to as low as 12 per cent could be obtained with some difficulty on less humid days with 5 persons in the room.

Male subjects between the ages of 18 and 21 years were selected after complete physical examinations by a practicing physician and mental classification by psychological tests [9] of general intelligence, resulting in the selection of a group of men of uniform intelligence and physiological make-up. Four of these subjects were used in most of the tests, while four men and four women of varying ages served as subjects in a few final tests in order to compare the results obtained with those of the trained subjects.

Preparatory to running the tests, the physiological-psychological conditions

in the psychrometric rooms were improved by replacing the old tungsten lights with five fluorescent lights in each room, giving a uniformly distributed illumination of 15 foot-candles. Small circulating fans were installed in each room, giving a uniform air movement of 24, $\pm$ 3, fpm throughout the occupied zone. Each room was equipped with an electric globe thermometer, to give the mean radiant temperature. In this new instrument, which is more fully discussed in another Laboratory report,[7] the globe-air temperature difference is determined directly by an 8-junction differential thermocouple with alter-

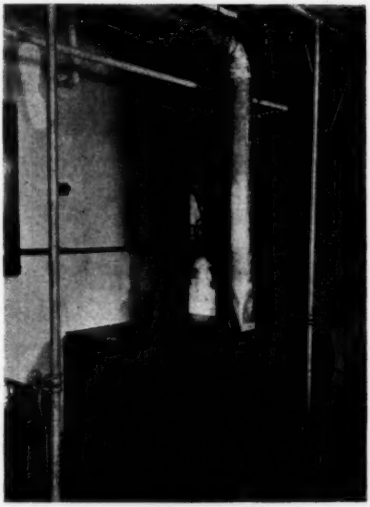

FIG. 2. VIEW OF AIR CONDITIONING EQUIPMENT USED IN TESTS

nate junctions in the globe and in the surrounding air. A Pierce hot-wire anemometer gave the air velocity near the globe.

The globe-air temperature differential varied throughout the tests from 0.3 to 1.3 deg. The observed variations in the mean radiant temperature ranged from a low of 0.5 deg above air temperature to a high of 2.4 deg above air temperature, and the average over all tests was 1.4 deg above air temperature. The mean radiant temperature is a measure of the environmental effect of radiation and implies the integrated average temperature of all surfaces within view of the point under consideration.

A recent Laboratory study[7] indicates that for a condition of 70 deg ET and 30 per cent relative humidity a 2-deg elevation or drop in the mean radiant temperature above or below the air temperature will about compensate for a 1-deg decrease or elevation in the effective temperature of the air. It will be observed, therefore, that the mean radiant temperature prevalent in the test room throughout this study should rarely affect the environmental condition

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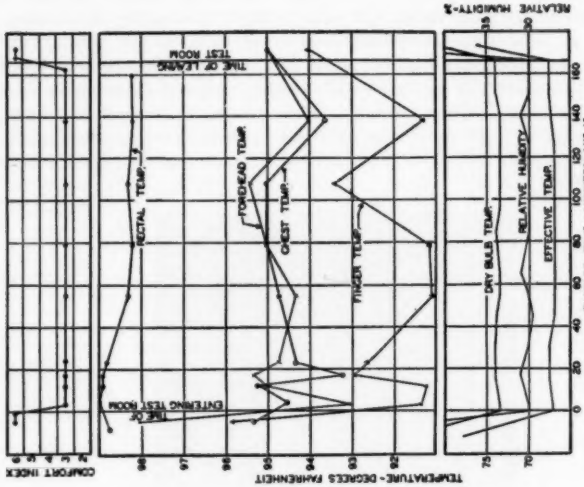

FIG. 4. LOG OF THE ATMOSPHERIC CONDITIONS MAINTAINED IN ONE TEST AND PHYSIOLOGICAL REACTIONS OF ONE SUBJECT WHILE COOL

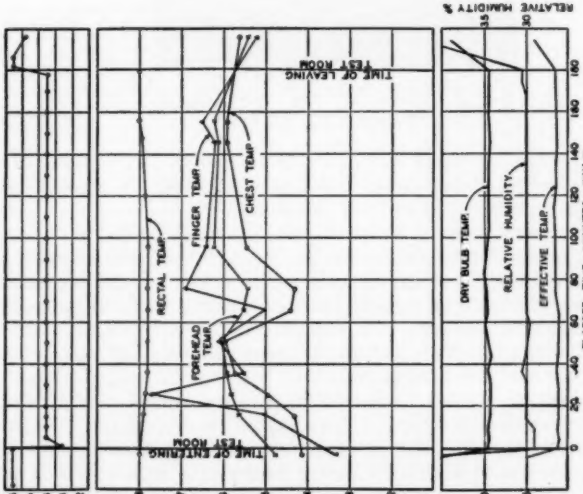

FIG. 3. LOG OF ATMOSPHERIC CONDITIONS MAINTAINED IN ONE TEST AND PHYSIOLOGICAL REACTIONS OF ONE SUBJECT WHILE IDEALLY COMFORTABLE

in the room by more than what would be had with a 0.7 deg ET change. Obviously, the mean radiant temperatures in the rooms were elevated by the higher skin and clothing temperatures of the occupants, by the lights, and by wall surfaces having slightly elevated temperatures due to the inflow of heat to the rooms. Since in this summer's study the room was always cooled below the surroundings, there were never any surfaces in the room in view of the globe that were at a lower temperature than the air.

The alterations in the psychrometric rooms and equipment served to im-

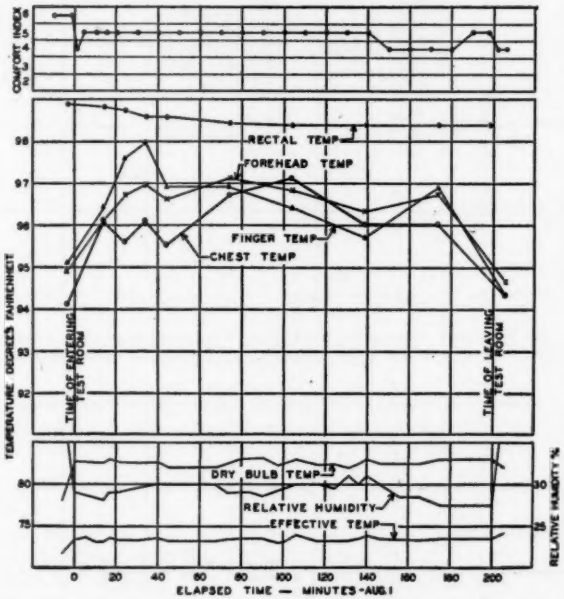

FIG. 5. ATMOSPHERIC CONDITIONS MAINTAINED IN ONE TEST AND PHYSIOLOGICAL REACTIONS OF ONE SUBJECT WHILE WARM

prove conditions therein by decreasing the *mean radiant temperature—air temperature difference* and by giving a more uniformly controlled air velocity. Since the globe thermometer means of measuring the mean radiant temperature has only recently been available it was not used in the earlier studies. However, a study of conditions within the rooms before these changes were made indicates that in former studies the mean radiant temperature may at times have been as high as 4 deg above the air temperature having an effect equivalent to a maximum of 2.0 deg ET.

TEST PROCEDURE

In conducting a given test, the rooms were conditioned to the desired effective temperature and relative humidity for a period of 3 hours before the subjects

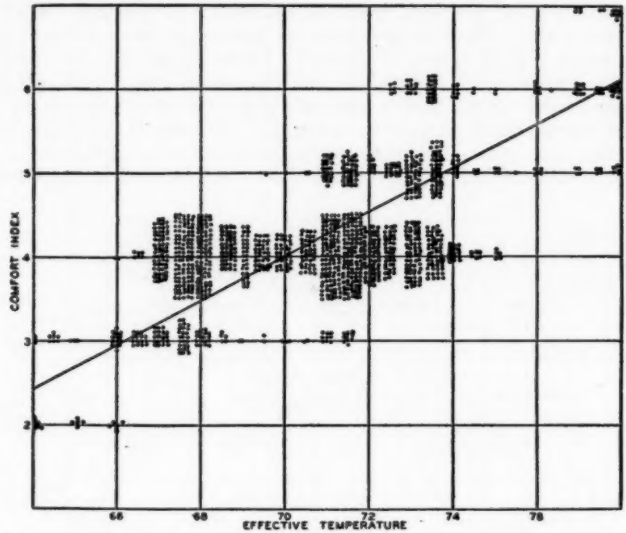

FIG. 6. RELATION BETWEEN EFFECTIVE TEMPERATURE AND COMFORT VOTES OF SUBJECTS IN TESTS AT 30 PER CENT RELATIVE HUMIDITY

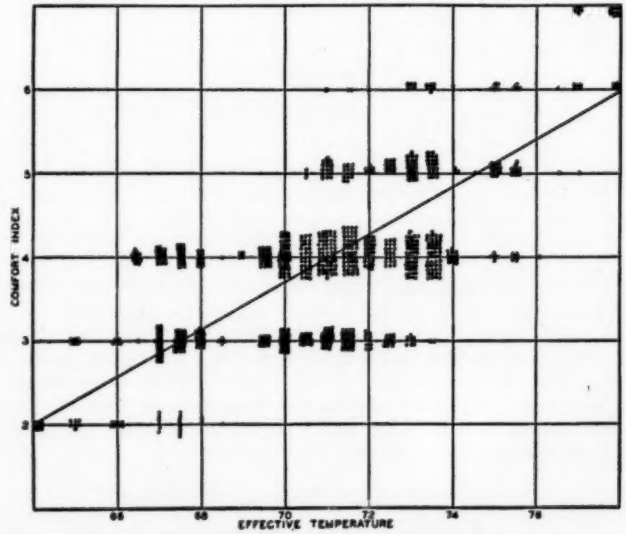

FIG. 7. RELATION BETWEEN EFFECTIVE TEMPERATURE AND COMFORT VOTES OF SUBJECTS IN TESTS AT 70 PER CENT RELATIVE HUMIDITY

entered from the prevailing outside condition, when the test started and continued for from 3 to 4 hours. Just before entering the test room, the pulse rate, degree of perspiration, feeling of warmth, body temperature, and the skin temperatures of the index finger, the chest and forehead of each subject were observed. These same observations were again made immediately after entering the room; the feeling of warmth first, and the other observations as rapidly as possible thereafter. Throughout the test period the feeling of warmth and the degree of perspiration of each subject were recorded every 15 min, while his other physiological reactions were determined at less fre-

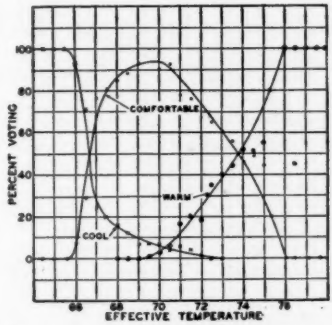

FIG. 8. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF SUBJECT VOTES FOR COMFORTABLE (4); FOR COOL (2 AND 3); AND FOR WARM (5 AND 6)

quent intervals. At the close of the test the subject again re-entered the outside, immediately reporting his feeling of warmth.

The subjects expressed their feeling of warmth in accordance with the index used in the earlier studies at the Laboratory; that is, *4* indicated a feeling of ideal comfort; *3* and *5*—comfortably cool and comfortably warm, respectively; *2* and *6*—cool and warm, respectively; and *1* and *7*—cold and hot, respectively. The degree of perspiration was observed in accordance with an index wherein *0* represented freedom from perspiration; *1* clammy; *2* damp; *3* surface entirely covered with perspiration or standing in beads or droplets; *4* perspiration runs and drips or soaks through clothing.

Skin temperatures were taken by means of No. 36 copper-constantan thermocouples mounted on cork buttons which were held on the skin so that the exposed junctions were in direct contact with the skin.

TEST DATA

Twenty-six tests were conducted at 30 per cent relative humidity with constant effective temperatures ranging from 64 to 78 deg for the different tests. A few of the early tests at 30 per cent relative humidity were made with the

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effective temperature ranging from a low value at the beginning of the test when the subjects were *cool* through a condition where they were *comfortable* to one where they were *warm*, after which the temperature was again slowly returned to the original condition. In other tests this cycle was reversed. These tests served to give a quick survey of the entire range but their results were not interpreted as giving the optimum condition with as great accuracy

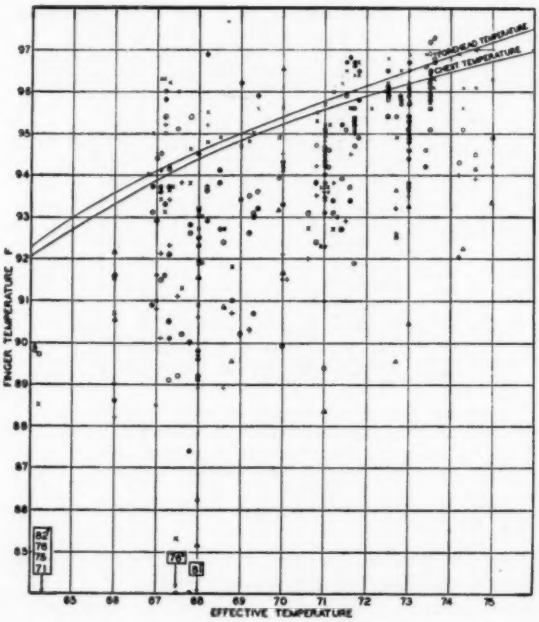


FIG. 9. RELATION BETWEEN EFFECTIVE TEMPERATURE OF ATMOSPHERIC CONDITION AND INDEX FINGER SKIN TEMPERATURES OF THE SUBJECTS. THE CHEST AND FOREHEAD TEMPERATURE CURVES FROM FIGS. 10 AND 11 ARE SUPERIMPOSED

as the greater number of tests in constant atmospheric conditions. A few tests were conducted with relative humidities as low as could be had (usually from 12 to 15 per cent) with four subjects and one observer in the room. Additional tests were made at 70 per cent relative humidity in an attempt to establish a control comparison with earlier Laboratory studies.

Logs of three tests on one subject when the conditions were respectively comfortable, somewhat cool, and somewhat warm for his comfort are given in Figs. 3, 4, and 5. Figs. 6 and 7 give the results of 26 tests at 30 per cent relative humidity and 13 tests at 70 per cent, respectively, while Fig. 8 gives

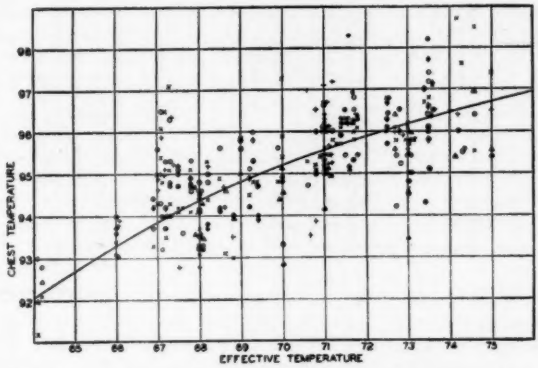


FIG. 10. RELATION BETWEEN EFFECTIVE TEMPERATURE OF ATMOSPHERIC CONDITIONS AND CHEST SKIN TEMPERATURES OF SUBJECTS

the relation between the effective temperature of the conditions in the different tests and the percentage of votes for *ideally comfortable*, for *comfortably cool* and *cool* combined, and for *comfortably warm* and *warm* combined. The relation between the effective temperature of the condition and the observed finger, chest, and forehead temperatures is plotted in Figs. 9, 10, and 11, respectively. Curves are drawn for the chest and forehead temperatures which for comparison are superimposed on the finger temperature chart.

DISCUSSION OF THE RESULTS

The logs of the three tests on one subject, Figs. 3, 4, and 5, show the same characteristics of data found in earlier studies. The careful observation of

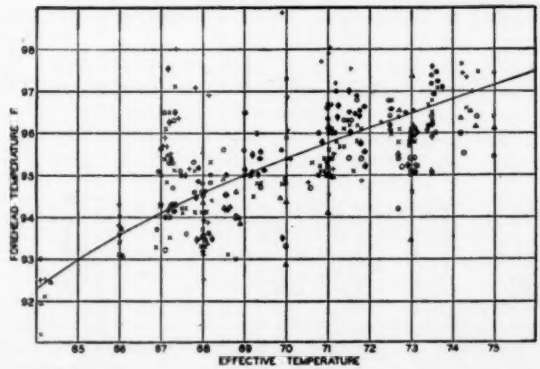


FIG. 11. RELATION BETWEEN EFFECTIVE TEMPERATURE OF ATMOSPHERIC CONDITIONS AND FOREHEAD SKIN TEMPERATURES OF SUBJECTS

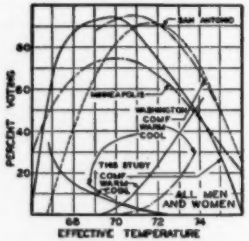


FIG. 12. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF IDEAL COMFORT EXCEPT WHERE OTHERWISE INDICATED. CURVES FOR THIS STUDY (30 PER CENT RH), WASHINGTON, MINNEAPOLIS, AND SAN ANTONIO

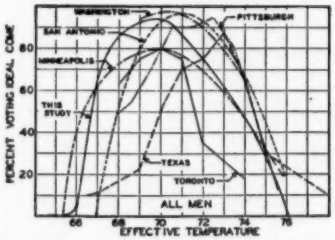


FIG. 13. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF COMFORT FOR ALL MEN FOR THIS STUDY (30 PER CENT RH), WASHINGTON, SAN ANTONIO, MINNEAPOLIS, PITTSBURGH, AND TORONTO

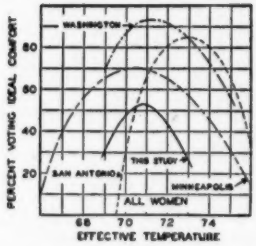


FIG. 14. RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF COMFORT FOR ALL WOMEN FOR THIS STUDY, WASHINGTON, SAN ANTONIO, AND MINNEAPOLIS

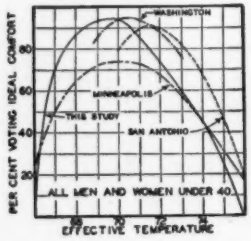
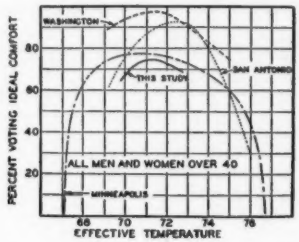


FIG. 15 (*Left*). RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF COMFORT FOR MEN AND WOMEN UNDER 40 FOR THIS STUDY (30 PER CENT RH), WASHINGTON, SAN ANTONIO, AND MINNEAPOLIS

FIG. 16 (*Right*). RELATION BETWEEN EFFECTIVE TEMPERATURE AND PERCENTAGE OF COMFORT FOR ALL MEN AND WOMEN OVER 40 FOR THIS STUDY, WASHINGTON, SAN ANTONIO, AND MINNEAPOLIS



body, and skin temperature of the forehead, finger and chest brought out one fact with greater conclusiveness than in previous studies—that is, when the subject was *cool* his index finger temperature dropped considerably below that indicated for his forehead and chest. This effect was quite universally indicated by the tests in this study. Whenever a subject had a feeling of coolness, particularly if this feeling persisted for some time, his finger temperature showed a decided drop. It was further observed by questioning the subjects and observing their data that this drop in finger temperature usually accompanied a more than casual sense of coolness and was expressed by the subjects as *a feeling of cold all the way through himself*, or as *feeling cold in his bones*. This observation is more or less in agreement with the findings of Sheard.[8]

The relationships between skin temperature and the effective temperature of the condition are better shown in Figs. 9 to 11. There is no apparent separating out of the skin temperature points for the different subjects indicated by the different symbols. Considerable variation is shown for the points in the three figures. They indicate in general, however, a drop in skin temperature with atmospheres of lower effective temperature. The relations for the chest and forehead skin temperatures are somewhat consistent and representative curves are drawn. These curves, superimposed on Fig. 9, show that while the chest and forehead skin temperatures do not differ greatly from each other they are both considerably higher and much less erratic than the finger temperatures. It will be noted that the finger temperatures became particularly erratic for cool atmospheres, occasionally falling to very low values as indicated by the points shown to fall below the minimum finger temperature ordinate of 84 F.

The data plotted in Figs. 6 and 7 show the usual characteristics for studies of this type and indicate optimum comfort at 70 deg ET with 30 per cent relative humidity, and at 71 deg ET with 70 per cent relative humidity. The plotting of the percentage votes for 30 per cent in Fig. 8 further shows optimum comfort to be at 70 deg ET with the usual falling off at lower or higher temperatures. The more rapid drop in the curve for low temperatures than is the case for high temperatures has been shown in earlier Laboratory studies to be characteristic.

The 70 deg ET indicated for comfort at 30 per cent relative humidity when compared with the optimum point of 71 deg ET for 70 per cent in this study might be construed as indicating that a slightly lower temperature is necessary for comfort at low humidities. If this were conclusively proved it would indicate that the slope of the effective temperature lines in this temperature range are slightly in error. Previous studies, as indicated in Figs. 12 to 16, show a certain degree of variation in the effective temperature required for optimum percentage of comfort in the different studies in different geographical locations on men and women of different age groups. The variations in effective temperature do not however correlate particularly well with any of these differences and it is quite possible that the one degree difference here indicated may not be significant. These different studies appear to show a more positive correlation between the effective temperature for comfort and the severity of heat of the summer. Thus, the low optimum temperature found in this study may possibly be accounted for by the less severe heat of the past summer. A log of the maximum daily temperatures observed by the Pittsburgh Weather

Bureau during the period included in this study is given in Fig. 17. It will be seen that a maximum of 90 F was exceeded only 15 times and that the highest maximum was 96 F. An analysis of the daily variation in the optimum condition desired for comfort in relation to the outside temperature showed no consistent variation.

The few tests with other than trained subjects, including men and women of different age groups, show the same general tendencies found in earlier studies as plotted in Figs. 12 to 16. Women and persons over 40 show a desire for slightly higher temperature.

The comparison of all curves in Figs. 12 to 16 adds further proof to the point brought out in a previous study [5] that with all the variations in geographical location, sex, and age, and now the addition of relative humidity, the variation in the effective temperature requirements for optimum comfort is relatively

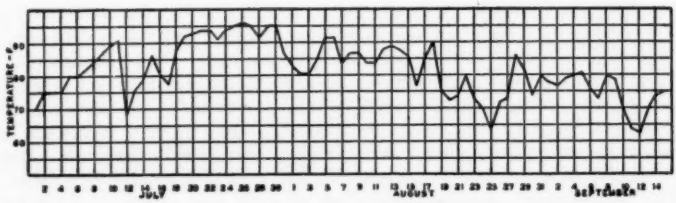


FIG. 17. LOG OF MAXIMUM DAILY TEMPERATURES REPORTED BY THE WEATHER BUREAU IN PITTSBURGH

small, ranging from 70 to 73 deg. The greatest single factor in this variation seems to be latitude or more likely prevailing summer conditions.

The curves in Fig. 18 giving, by their intersection with the *comfort 4* ordinate, the effective temperature for optimum comfort, and by their slopes the change in the effective temperature necessary to give a single index change in the feeling of warmth, for various studies made by the Laboratory since 1923, including this study, are of interest. They show the variations in effective temperature required for comfort not only with summer cooling and air conditioning but also with winter heating and air conditioning. With the exception of the first study made in 1923, optimum comfort is indicated to range from 67.5 to 73 deg ET. Also, with the exception of the first study, the slope expressed in degrees effective temperature for a unit change in the comfort index indicates that from 2.3 to 4.3 deg ET change is required for a unit comfort index change or a change from *ideally comfortable* to *comfortably warm.*

DRAFTS, SHOCK AND OTHER PECULIAR SENSATIONS

Analyses of the reactions to draft, sense of coolness, and other local sensations upon entering the rooms show no noticeable variation from earlier studies. No indications of a draft were recorded among the 1,627 times which the individual subjects were asked to record their reactions. This may be accounted for by the better control of air velocities in this study. An analysis

of cold shock experienced upon entering the conditioned space shows no tendencies different from those found in earlier studies. A decisive feeling of coolness was experienced by the subjects upon entering the cooled space from the hot outside, particularly when the body was wet with perspiration. The severity and period of these experiences, however, show no measurable differences from those found in earlier studies, and the subjects usually indicated that the feeling of coolness was a pleasant rather than an unpleasant sensation.

The lower relative humidity at which this study was conducted led to including among the questions asked of the subjects those of indications of dryness of the lips, throat, or nose, as well as dryness of the skin. There were four indications of dryness of the lips, and none of the throat, nose, or skin recorded among the 1,670 interrogations, indicating conclusively that no such sensations predominated at a relative humidity of 30 per cent. In the few tests conducted at relative humidities ranging from 12 to 15 per cent, however, there was a fairly decisive increase in such observations of dryness, usually applying to the lips and nose.

The warm shock experienced upon re-entering the hot outside after spending several hours in the cooled and air conditioned space showed no measurable difference from that found in earlier studies at higher relative humidities. There is considerable conjecture concerning the reasons for either the cold shock upon entering a cooled space or the warm shock upon leaving. Assuming that the air in a cooled space has both its temperature lowered and its moisture content decreased below that prevalent in the outside atmosphere, there are two factors affecting either shock sensation—namely, the temperature and the moisture content (or vapor pressure) of the air, in both the conditioned space and the outside. Before entering the cooled space, a person's body will be as nearly in equilibrium with the hot environment as can be realized by the ability of life to adapt itself to a given atmospheric condition. If the effective temperature is high for comfort, the body will be wet with perspiration, thereby encouraging the proper rate of evaporation of perspiration to eliminate the excessive body heat resulting from the prevailing temperature and vapor pressure of the atmosphere. Immediately upon entering a cooled space the body will not be in equilibrium with its environment—the availability of moisture on the body is excessive for the new environment and the vapor pressure of the atmosphere is lower, both factors encouraging more rapid evaporation and therefore a higher rate of heat loss from the body. The fact that the conditioned atmosphere has the lower dry-bulb temperature will in some measure reduce evaporation for the reason that evaporation always requires the availability of heat from the surroundings—in this case, from the atmosphere and the body. The combined result of these three changes will be such as to increase the rate of evaporation until equilibrium has been re-established through a decrease in the amount of perspiration available. While the body, no doubt, very quickly exercises control in reducing the rate of perspiration a certain time must necessarily elapse before the moisture already on the body surface is reduced to equilibrium. Hence, while the sensible heat loss increases upon entering the conditioned space because of the greater *body surface temperature —air temperature* difference the latent heat loss also increases, resulting in an excessive feeling of coolness.

Upon re-entering the warm outside after the body has reached equilibrium with the inside, the reverse takes place. Within the conditioned space the

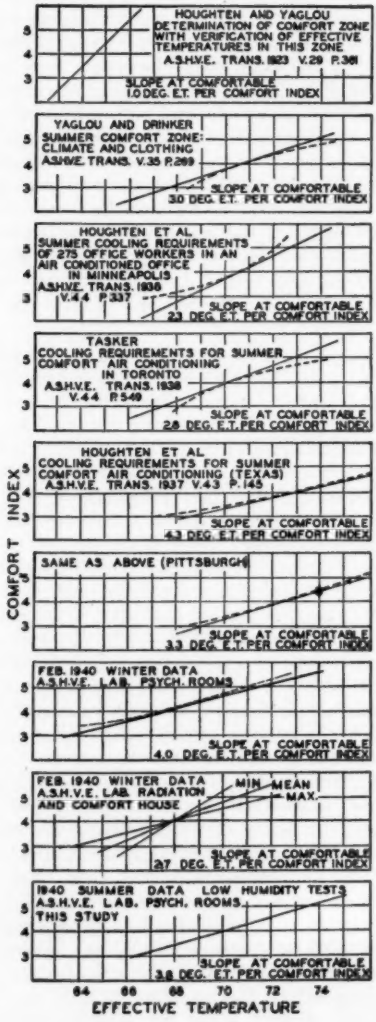


FIG. 18. RELATION BETWEEN EFFECTIVE TEMPERATURE AND OPTIMUM COMFORT FOR SUBJECTS IN VARIOUS STUDIES SINCE 1923

body keeps latent heat loss down by limiting the availability of perspiration for evaporation, so that the combined sensible and latent loss results in equilibrium between the rate of heat production and heat loss. Upon entering the hot and humid outside, sensible heat dissipation is immediately reduced, and because of the higher vapor pressure of the atmosphere and the freedom from perspiration on the body, latent heat loss must also be reduced. As a result the person will have an excessive feeling of warmness until the body has again re-established the availability of perspiration on the surface so that in the new surroundings the latent loss will increase to re-establish a balance between total heat production and dissipation. Since availability of perspiration on the surface in the most important of the three factors, however, it is probable that perspiration on the surface before entering the cooled space and the lack of perspiration on the surface upon re-entering the warm outside conditions account for the greatest effect in the feeling of coolness upon entering and the feeling of warmness upon leaving the conditioned space.

CONCLUSIONS

While the study indicates a possible lower requirement of 1 deg ET at 30 per cent relative humidity when compared with that at 70 per cent, this difference is barely significant, and the relatively low optimum effective temperature indicated may be attributed to the prevalence of relatively cool weather conditions during the past summer.

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7. ASHVE RESEARCH REPORT No. 1172—Radiation as a Factor in the Sensation of Warmth, by F. C. Houghten, S. B. Gunst, and John Suciu, Jr. (ASHVE TRANSACTIONS, Vol. 47, 1941.)

8. ASHVE RESEARCH REPORT No. 1110—Skin Temperatures of the Extremities

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and Effective Temperature, by Charles Sheard, Ph.D., Marvin M. D. Williams, Ph.D., and Bayard T. Horton, M. D. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

9. Otis Self-Administering Tests of Mental Ability, by Arthur S. Otis. (World Book Co., Yonkers, N. Y., 1922.)

DISCUSSION

THOMAS CHESTER (WRITTEN): This is a useful paper and its substantial confirmation of effective temperatures at fairly low relative humidities shown by the Society's Comfort Chart, reflects credit on the early research work antecedent to the appearance of the Chart.

The stated possibility that a lower effective temperature of 1 deg at 30 per cent relative humidity is desirable in comparison with the value at 70 per cent, can no doubt be attributed to the slightly reduced heat loss by radiation from the subjects to the somewhat warmer walls. This seems more reasonable than the theory advanced that the discrepancy can be ascribed to the subjects being acclimated to a relatively cool summer.

For the same effective air temperature, the room boundary inner surface temperatures must be warmer with low relative humidities than with higher relative humidities, because the dry-bulb air temperature is higher.

The remarks in the paper concerning entering and leaving shocks are very good and unquestionably will improve our understanding of the sensations and reactions involved.

Now that globe thermometers are available, it seems desirable that further research work should be carried on in possibly a hotter summer, and with relative humidities below 30 per cent in the experimental rooms. There is scope for addition to our knowledge of the effects of variations of heat disposal from the human body by radiation, as affecting comfort.

E. P. HECKEL (WRITTEN): This paper based on the results and studies at the ASHVE Research Laboratory is most interesting.

While the results as shown amplify and quite definitely confirm the correctness of previous studies made in the higher relative humidity brackets, we find a new note of interest and one bearing on a problem with which many of us have been confronted at various times. I refer particularly to the question of cause for discomfort, frequently referred to as shock, which is occasionally experienced upon entering or leaving air conditioned spaces.

The paper states, and I believe correctly, that "there is considerable conjecture concerning the reasons for either the cold shock upon entering a cooled space or a warm shock upon leaving." It is my opinion, based upon observations and analyses made of various indoor and outdoor conditions at the time of such experienced discomforts, that the air moisture content difference (vapor pressure difference) between the inside air conditioned space and the outside atmosphere is the contributing cause of momentary or prolonged discomfort, requiring a physiological readjustment when going from one condition into another as referred to in this paper.

While the dry-bulb temperature difference of 15 deg. between inside and outside conditions (95 deg-80 deg) may be considered a factor in causing such discomfort, it is believed that the temperature difference when considered alone is a negligible factor. Neither can we correctly tie-up the cause to the difference in relative humidities as being the prime direct factor, since we frequently experience sensations of discomfort when going from an inside conditioned space of 80 deg and 50 per cent relative humidity, to the outdoors, having a dry-bulb of 90 deg and 50 per cent relative humidity. Note the relative humidity is the same inside and outside.

When emerging to an outside atmospheric condition of 90 deg dry-bulb with 40 per cent relative humidity, from an inside condition of 80 deg dry-bulb and 50 per cent relative humidity, we feel comparatively little or no shock effect. The dry-bulb temperature difference referred to inside and outside is the same in both cases; but in the first case there is a dew-point temperature difference of (69 deg-60 deg) 9 deg which, expressed in terms of air moisture content difference, is (107-77) or 30 grains per pound of air, whereas in the second case, free from shock effect, the dew-point temperature difference is only 2.5 deg or, as expressed in terms of air moisture content difference, (80-77) 7 grains per pound. The greater the difference between inside and outside air moisture content, the greater the sensation of momentary chilling or the effect of vapor pressure smothering, depending upon whether you are passing from a lower to a higher environment, or vice versa.

The greatest effect of discomfort that I have experienced (aside from entering a steamed room) was produced by emerging from a conditioned space of 80 deg dry-bulb, 50 per cent relative humidity, to the outside atmosphere having an 84 deg dry-bulb with 76 per cent relative humidity, indicating a dew-point of 76 deg, as contrasted to 60 deg dew-point in the conditioned space. As can be checked from the psychrometric chart the difference in air water vapor pressure was 0.3828 in. of mercury and the air moisture content difference was 136-77 or 59 grains per pound.

Obviously I concur with the paper as presented, that the moisture content difference or, if you prefer, the vapor pressure difference, is the important contributing factor in the cause of shock discomforts, upon entering or leaving cooled spaces.

A. B. NEWTON (WRITTEN): Surely the authors of this paper have commenced a much needed extension of our knowledge of desirable temperature and humidity ranges.

The details of the two psychrometric rooms are of great interest and there is much to be learned here in setting up future test programs. I am wondering whether use of fluorescent lighting to reduce the mean radiant temperature is completely desirable, since on most actual field installations the warm walls, sun effect, and standard lighting all tend to raise the radiant temperature some 2 to 4 deg above the air temperature.

In practice I seem to note a wider effective temperature tolerance at low relative humidity than at high, and particularly the tolerance narrows down as we approach 65 or 70 relative humidity. I am therefore interested in a comparison of Figs. 6 and 7 of this paper.

Fig. 6 shows a small slope of the comfort index line at 30 relative humidity which is in line with field observations and the test points obtained in the laboratory. However, Fig. 7 does not indicate the steepening of this line we customarily find in practice at the higher relative humidity values. I would suggest that a steeper curve could be drawn starting about at index 2 at 67 deg ET and running to index 6 at 75 deg ET.

Such a line seems to represent the test data fully as well and yet agrees more fully with what we find in our field installations.

In the comparisons between this study and previous ones in Figs. 12 through 16, is it not possible that the peaks of the curves for this study are thrown somewhat to the right, comparatively, due to the use of fluorescent lighting and the resulting lower mean radiant temperature? This might make a degree or two difference in the optimum temperature, and if such an effect is present it could mean a definite requirement for lower effective temperature at low relative humidity.

We still have a great deal to find out to explain comfort reactions we obtain in our test work, and one of our greatest problems seems to be that of getting laboratory tests effectively to simulate field conditions. Particularly in the last two

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years or so, we have had indications that the cyclical variations in temperature and humidity which occur from starting and stopping of equipment are more important than we once thought. Many jobs experience a variation of 2 deg dry-bulb and 10 to 15 per cent relative humidity from two to three times an hour. Often on the simpler jobs this cannot be avoided. Has any of the laboratory work indicated the practical tolerances to such variations, and can we state accurately the magnitude of such variations experienced in the present tests?

Surely the good correlation we are obtaining in scattered test programs correlates very nicely, and indications are that effective temperature will remain an excellent index of comfort.

J. DEB. SHEPARD (WRITTEN): The authors, in their series of tremendously important investigations on the subject of effective temperatures, have done a remarkable job of reducing human beings to scientific instruments for the purposes of our research. However, as a result of the meticulous care with which the problem has been approached as a scientific study, I feel that we lose sight of the fact that nature has fortunately created human beings with physical equipment which permits the body to adapt itself to a remarkably wide range of atmospheric conditions. Out of this wide range of conditions, we ask the body to tell us only what feels alike from the strict sense of sensation of heat or cold, and the body answers the conditioned mind within the disciplined limits of the question.

On the practical side, however, people do not buy air conditioning on this basis. They pay money to give their bodies something that justifies the price in terms of pure bodily satisfaction and, as a result of my limited observations, I am sure that if given a free choice bodies give a happy answer to minds when they encounter moderate temperature and low humidities.

Since the ultimate aim of all our research is to sell more air conditioning, the creation of conditions resulting in the greatest pure body satisfaction best satisfies both our altruistic and practical aims. A happy body is probably a healthy body, and is also a good house for a healthy mind; and a happy body tells a healthy mind, *this is good—let's buy it and pay for it.*

These thoughts lead me to ask that we have more research on this subject along the lines of the excellent field studies that have been made in Toronto, Minneapolis and San Antonio, but with less emphasis on *is it warm* or *is it cold?* and more on *how do you like it?* If we know what makes the public answer *swell,* to this latter question, we will build a great industry and offer a service which will make our great country greater!

PRESIDENT FLEISHER: I want to take exception to the curves which are superimposed on the findings of this particular paper, as shown in Fig. 6, especially when you are examining individuals. The probabilities are that the same individual felt the same experience in Fig. 6 all the way from 67 to 74 and you really cannot get statistical with individuals.

The consequence is, that the line on Fig. 6 should be a horizontal and not a diagonal line, rising at the beginning of the chart and falling off very rapidly at the end. Certainly there is no reason as indicated by any of these investigations to show that the curves, except for the composite numbers, should go in any particular direction and no individual likes to consider himself as composite. Consequently, there is probably no excuse for the line showing in that particular direction.

We have boundaries, and between boundaries we have for a great distance horizontals. As I say, we are not going to anticipate our own paper‡ except to say according to the theory of my son and myself that the two conditions supposed to be different, as investigated by Mr. Houghten, that is, 71 deg ET with 70 per cent

‡ Comfort and Health and Temperature—A Mathematical Solution, by W. L. Fleisher and W. L. Fleisher, Jr., see p. 175.

relative humidity, and 70 deg ET with 30 per cent relative humidity, we think are exactly alike in their effect.

L. T. AVERY: Maybe the purpose of our test work is to sell air conditioning, I do not know. There is one question I would raise in the light of our own experience. Possibly we have a reason for thinking as we have in the past, that lower humidities were a little more comfortable. We happen to have, in one building, several systems whereby, because of well water cooling we have a high humidity in another. Those were put in at the request of the engineer and the owner feels he prefers the lower humidity.

I give you two points that differ widely from this paper. We do not carry 71 deg ET, but 73 deg ET. People are not sitting at rest, they are in motion in office work, moving, working at desks, etc. Now, maybe there is something there that escapes the eye. The other thing we have observed is that with high humidities there is a very distinct increase in odors, stale odors of the conditioner itself, possibly due to lack of cleanliness on the coils or filters. Perhaps the increase in odors comes from carpets, leathers, or draperies. Accumulated tobacco odors are particularly bad in high humidity. Perhaps there are other things besides temperature and humidity and does it feel *equally cool,* when it comes to saying what is the *ideal* in air conditioning.

W. F. RYAN: I wish to corroborate with Mr. Avery in his statement. Tests we have made in portions of dry Kansas from which I come, coincide with Mr. Avery's remarks.

C.-E. A. WINSLOW: This is an excellent piece of work, and adds a great deal to our knowledge. One point, I would like to call attention to, the last curves shown by Mr. Houghten to establish as comfortable a much higher temperature than the old 66 F. I think the reason is, the cold radiation factor. Radiation to cold walls in the early experiments was entirely eliminated and air movement relatively low.

I should like also to underline what Mr. Fleisher said about the charts. These charts are shown with a single line, and I judge they are plotted by eye. It seems to me this would be better in analyzing data which have a good deal of scatter to determine the line mathematically, and then take a reasonable probable error either way and plot the limits of the zone within which the values have a probable significance.

I think the scatter is too great to match them against a single line. Instead of a line drawn by eye, one might better show a band within which 75 per cent of the observation fall between the upper and lower limits of that band.

MR. HOUGHTEN: The only question that requires an answer is that raised by Mr. Fleisher, and also by Dr. Winslow.

It is agreed that the test points scatter. Where the points scatter so badly, the curve drawn through them does not mean much. All it can mean is an average. However, the second point raised by Mr. Fleisher in this connection, while bearing some logic, may also be argued from the other side. We are speaking of Figs. 6 and 7, where the maximum comfort line passes through a series of points. It is argued that these are not the points of individual reactions. While this is true it would seem to be the only logical procedure to be followed, and it is the same procedure as that followed in the studies* made at Minneapolis, Washington, San Antonio, and particularly the one made in New York by Dr. W. J. McConnell.

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* See Bibliography, (6).

In this study there were only a few subjects. Unfortunately, the difficulties and cost involved in obtaining subjects for studies of this kind does not permit the use of a large enough number to give sufficient points for a statistical analysis.

The Minneapolis study included 275 individuals. Yet the results when plotted showed no noticeably greater spread than in this study. Dr. McConnell's study in New York was made on something over 2,000 workers in the Metropolitan Life Insurance Co. offices. Yet here again the spread of points was not noticeably greater.

No. 1175

THE INFLUENCE OF PHYSIOLOGICAL RESEARCH ON COMFORT REQUIREMENTS

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This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Illinois, College of Medicine.

MEMBERS of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS have proposed to the public that they can furnish a controlled atmosphere, which is preferable to the uncontrolled one supplied by nature. This proposal sets up a thesis with two major premises, and shoulders the responsibility for their proof upon the Society. In the first place the atmospheric environment must demonstrate its value by bringing more comfort to its subjects than an uncontrolled one. If it does this, people will spend money to secure it. In the second place, it must be so selected and so secured that harm will not result to the subjects. Both of these premises were early recognized by the Society and investigative work was undertaken to define the comforts and dangers of conditioned air.

For the purposes of discussion of the problems involved, a crude but simple analogy may be adopted. If an incubator, which has varying quantities of heat liberated within it, is furnished with a delicately operating thermostat, its temperature should be maintained at a relatively constant level. If this incubator is now placed within a room whose temperature is controlled within 3 to 4 F—the maintenance of a constant temperature will prove easy. If it is placed out of doors the task will become exceedingly difficult.

THE VARYING HEAT PRODUCTION

The transformations of energy within living cells as measured by heat production are dependent largely on the temperature of the cells. In certain animals (lizards, frogs, the so-called poikilothermous animals) a mechanism for the maintenance of a constant temperature is absent or poorly developed. When the surroundings are cold, the rate of heat production falls, activity

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Presented at the 47th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., January, 1941.

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slows, and at times apparently ceases. The animal suspends animation and hibernates. When the sun shines again and warms up the cells, oxidations speed up and activity appears. The transformations are governed almost entirely by the environmental temperature. Work can be accomplished only when the temperature reaches an optimal level permitting the cells to draw upon stored sources of energy. Once this energy is converted there is no mechanism for the conservation of the extra heat developed.

In man a heat regulating mechanism for the maintenance of a constant temperature of 98.6 F has been developed. Any departure of the temperature from this point interferes with the energy transformations. The cells cease to function and the individual becomes seriously crippled or dies. The upper endurable limit of temperature over a limited period of time is found around 106 F. The lower limits have been extended somewhat, recently, by the refrigeration experiments of Temple Fay.[1] In health the body temperature remains surprisingly constant through all varieties of environmental temperatures and this has extended the functional capacity of the biological machine in man to 12 months out of the year. A constant supply of energy is required to maintain this temperature level of the body and to keep it in a state of preparedness for work. This exchange is spoken of as the basal metabolism and its magnitude is dependent on the sex, age and surface area of the individual. The heat production takes place in all the active cells. Loewy[2] estimates that the heart action accounts for about 3.6 per cent, and the respiratory movements for 10 per cent. Krogh[3] places the kidney requirements at 5 per cent, and those for the functional activities of various organs at 25 per cent. This leaves 75 per cent of the heat to be produced by oxidations in the tissues apparently at rest. The muscles represent more than half of the total weight of the soft tissues of the body and it is in these that the great bulk of the heat is developed. This is substantiated by the rapid fall in the heat production which follows the isolation of the muscles from the nerves with a drug (curare) or by large doses of anaesthetics. This basal energy exchange may be reduced by 40 per cent if the individual is deprived of his thyroid gland and by 20 per cent if the adrenal glands are damaged. This leaves a residual heat production which would have to be attributed to the temperature at which the cells were maintained.

If an individual is given food the heat production is increased approximately 10 per cent. If he is required to do work, it may be increased 500 per cent or more. The mechanical efficiency of the body is usually placed at 20 per cent, so that the heat required for a small amount of work is large. One may now visualize the daily heat production as a phasic process. In the early morning hours the machine is idling and the heat production is at a basal level. Breakfast is eaten and work begun. The production now rises 100 to 150 per cent. After lunch with a period of relaxation the heat production falls. So throughout the day, the energy uptake fluctuates widely. Despite these fluctuations the body temperature is guarded so jealously that it does not vary significantly. In view of the great dependence of the poikilothermous animals on heat production to maintain body and cellular temperature, the question naturally arises

[1] Temperature Factors in Cancer and Embryonal Cell Growth, by L. W. Smith and T. Fay (*Journal American Medical Association*, 113:653-660, 1939.)
[2] Basal Metabolism in Health and Disease, quoted by E. F. DuBois. (*Lea and Febiger*, Philadelphia.)
[3] Loc. Cit. Note 2.

as to what extent man and the homeothermous animals make use of variations in heat production to accomplish this objective.

ROLE OF HEAT PRODUCTION IN REGULATION OF BODY TEMPERATURE

It can be seen that, if one is to answer this question, experiments should be conducted with the subject at a basal level of energy expenditure. All the energy developed by the machine is being used to supply its own inherent demands, of which temperature maintenance is a vital one. McConnell, Yaglou and Fulton,[4] working under the auspices of the ASHVE, reported in 1925 on experiments covering basal metabolism before and after exposure to high temperature and various humidities. Their subjects were in the basal state (deprived of food and water for 12 hours) and were lightly clothed. The oxygen consumption was measured in the comfortable primary room and again in the warm experimental or secondary room. The values were plotted against effective temperature. It was found that below 65 deg ET and above 85 deg ET, the oxygen consumption increased. They found a zone of minimum metabolism between 75 deg ET and 83 deg ET. Above 95 deg ET there was an increase in body temperature. In this zone it is obvious that conditions were not basal and that the subject's metabolism had increased because of the higher temperature of the metabolizing cells. Wiley and Newberg[5] studied one male subject under basal conditions over an effective temperature range of 59.9 to 84.3 deg with a relative humidity of 20 per cent. They found that the heat production was practically constant between 68.7 deg ET and 81.7 deg ET. In 1937, Hardy and DuBois[6] studied nude male subjects in a calorimeter. Since the relative humidities in the calorimeter experiments varied between 22 per cent and 47 per cent with an average of 30 per cent, the effective temperatures have been plotted on the basis of a 30 per cent relative humidity. They found the heat production constant between 67 deg ET (22.7 C) and 78 deg ET (35 C). In the lower ranges studied, the constant heat production was augmented intermittently by chilling. The chilling could be postponed for a time by a voluntary inhibition of the shivering reflex. It could not, however, be prevented in the face of an increasing heat deficit.

It is well understood by all, that chilling may be abolished by active voluntary movements as swinging the arms, walking vigorously, or other similar movements. In 1940, the same investigators[7] reported that between 72.0 deg ET (27 C) and 74.5 deg ET (29 C), two out of three female subjects showed a drop of metabolism of 14 per cent below that of men. In this respect women would seem to differ from men. In 1938, Hick, Keeton and Glickman[8] reported studies on four hospital normal female subjects, clothed in light cotton pajamas. No significant changes were noted in oxygen consumption between 66.0 and 85 deg ET. When the rectal temperature rose indicating a retention of heat, the basal production also increased. Similar studies were made in 1929

[4] Basal Metabolism Before and After Exposure to High Temperatures and Various Humidities. by W. J. McConnell, C. P. Yagloglou and W. B. Fulton. (ASHVE TRANSACTIONS, Vol. 31. 1925.)

[5] Journal of Clinical Investigation. by F. H. Wiley and L. H. Newberg. (10:689, 1931.)

[6] Regulation of Heat Loss from the Human Body. by J. D. Hardy and E. F. DuBois. (*Proceedings of National Academy of Sciences*, 23:624-631, 1937.)

[7] Difference Between Men and Women in Their Response to Heat and Cold. by J. D. Hardy and E. F. DuBois. (*Proceedings of National Academy of Sciences*, 26:389-398, 1940.)

[8] ASHVE RESEARCH REPORT No. 1079—Physiologic Response of Man to Environmental Temperature, by F. K. Hick, R. W. Keeton and Nathaniel Glickman. (ASHVE TRANSACTIONS, Vol. 44, 1938.)

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at a somewhat higher metabolic level by Houghten, Teague, Miller and Yant.[9] In these the subjects were allowed to have their usual breakfast and lunch. The experiments were begun at 2 p.m. The subjects were seated most of the time, but were allowed some latitude in their movemnts. Their metabolism was to be regarded as a standard one, which by calculation was adjusted to a level 36 per cent above basal requirements. They were exposed to varying ranges of relative humidity (20, 45, 70 and 95 per cent), in still air and in air

TABLE 1—THE ENVIRONMENTAL RANGES OF CONSTANT METABOLISM

INVESTIGATOR	EFFECTIVE TEMP. F.	OPERATIVE TEMP.	REMARKS
1. McConnell, Yaglou and Fulton	65.0–85.0	...	Basal and Lightly Clothed
2. Wiley and Newburgh	68.7–81.7	...	Basal and Nude
3. DuBois and Hardy (Men)	67.0–81.5	...	Basal and Nude (Did not extend investigations above 81.5 deg)
4. DuBois and Hardy[a] (Women)	67.0–72.0	...	Basal and Nude
5. Hick, Keeton and Glickman	66.0–85.0	...	Basal and Very Lightly Clothed
6. Houghten, Teague, Miller and Yant	66.7–83.7	...	Standard Metabolism = Basal + 36 per cent Normally Clothed
7. Winslow, Herrington and Gagge	...	64.4–105.8	Standard Metabolism = Basal + 16 per cent and Basal + 29 per cent Unclothed

[a] Starting at 72 deg ET the heat production falls and reaches a value 14 to 20 per cent lower at 76.9 deg ET, then remains constant up to 81.5 deg ET.

velocities of 235 and 385 fpm. The heat production was constant within the range of 66.7 and 83.7 deg ET. Winslow, Herrington and Gagge[10] reported studies on two subjects, unclothed and in the semi-reclining position, who had eaten a light breakfast or light luncheon 1 to 1.5 hours previously. The metabolism was characterized as *standard* but not basal. The values for Subject 1 were 16 per cent and for Subject 2, 29 per cent above the basal level. The heat production was found to remain approximately constant over the ranges of *operative temperatures* (64.4 to 105.8 deg) studied. Although the relative humidity varied between 40 and 50 per cent the environmental operative temperatures cannot be expressed as effective temperatures in all cases, because there were large experimental differences between the temperature of the walls and the surrounding air.

Observations of various investigators on the environmental range within which the metabolism is constant are summarized in Table 1.

[9] Thermal Exchanges Between the Human Body and Its Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. F. Miller and W. P. Yant. (*American Journal of Physiology*, 88:386, 1929.)

[10] Physiological Reactions of the Human Body to Varying Environmental Temperatures, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (*The American Journal of Physiology*, 120:1-22, 1937.)

One should at this point distinguish between basal heat production and extra heat produced intermittently by shivering or by voluntary activity. The basal heat production is a function of cell temperature, concentration of thyroxin and other calorigenic hormones. This is an unconscious source of heat and rarely reaches the attention of the individual unless there are gross abnormalities. From the evidence cited, it is obvious that the body does not accelerate or depress this unconscious type of heat production when extra heat is required. It is a fixed quantity through all the ranges of environmental temperature, in which the cellular temperatures of the body remain constant. When the nude body is exposed to below 74.2 deg ET over a relatively short period it cools and continues to lose heat to the environment without producing any extra heat until 66 deg ET [11] or lower is reached. At this point there is extra heat produced by shivering. A conscious and uncomfortable method of heat production is called into action. In ET's above 83.7 heat is retained. The temperature of the cells is increased and basal oxidations are increased. This process also is associated with discomfort, although the rise in cellular temperature per se is probably not the cause of the discomfort.

HEAT ELIMINATION

As previously noted, heat is produced within the interior of the body and eliminated from the surfaces. It must first be transported to the surfaces. The mechanism involved is largely a physiological one and its discussion will be continued later in the paper.

Once the heat reaches the surfaces, its dissipation is dependent on the physical properties of the environment. This is a physical problem and falls within the realm of the engineer. It may be helpful at this point to return to the analogy of the incubator and its environment. We have the choice of studying the loss of the heat from the incubator, when it is placed in the relatively uniform environment of a room with a controlled temperature, or when it is placed in the non-uniform environment of the outdoors.

LOSS OF HEAT FROM BODY SURFACES

For simplicity, experiments which have been conducted under less complicated thermal environments will be discussed first. Houghten, Teague, Miller and Yant [12] used an experimental room in which the air was relatively still. The room, air, and walls were approximately the same temperature and the humidity was known. The heat production of their subjects as shown in their data was 36 per cent above the basal level. The subjects were clothed. They were able to calculate the total heat loss by measuring the heat produced and adding to this the heat stored in the body positively or negatively as shown by changes in body temperature. They were also able to partition the heat loss due to evaporation, and by difference between this and the total heat loss, that due to radiation plus convection. DuBois and Hardy [13] measured in their calorimeter the total heat loss. They measured separately the heat loss due to evaporation and radiation, and arrived at that due to convection by difference. The heat production was at basal levels. Subjects were both nude and clothed.

[11] Loc. Cit. Note 6.
[12] Loc. Cit. Note 9.
[13] Loc. Cit. Note 6.

Their experiments are most important ones, since they give a clear picture of the temperature regulation of man under the simplest experimental conditions. For the nude subject they define a neutral zone extending from 83 F to 88 F (73.2 to 76.9 deg ET) ‡ in which the heat production and elimination balance each other with great exactness and ease. There is no progressive cooling of the body, no rise in body temperature indicating storage, and no sweating required to eliminate the heat. The sudden ingestion of cold drinks may push the individual into the cold zone resulting in fall in body temperature. Hot drinks move him into the warm zone and cause sweating to prevent a rise of temperature. When the subject is clothed, the neutral zone extends over 16 F with an extension into the cold zone, and lies between 64.8 and 76 deg ET. With heavier clothing it is obvious that this zone could be extended further in the cold zone.

It is now of interest to examine the comfort zones experimentally determined by F. C. Houghten and C. P. Yaglou [14] and by C. P. Yaglou and Philip Drinker.[15] The average winter comfort zone lies between 63 and 71 deg ET and the average summer zone between 66 and 75 deg ET. Under the conditions of the experiments, the subjects were clothed and were active to the extent that they were seated and exhibited a limited amount of voluntary movements. It is safe to estimate that their metabolism approximated a level of 36 per cent above basal. The note attached to the comfort chart is significant for the limitations under which the chart is applicable:

> Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort lines is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores and the like, where the exposure is less than 3 hours.

The limitations are obviously set up with the hope of reducing the environment to an approximately uniform one, and they require that the individual remain in it long enough to come to an equilibrium or a *steady state* to use a term employed by many physiologists. In this respect the conditions of comfort zone experiments approximate those of DuBois and Hardy and allow one to study the two together.

The *neutral zone* of DuBois and Hardy [16] for the clothed subject at basal conditions is between 64.8 deg ET and 76 deg ET [17] and corresponds very closely to the summer comfort zone 66 deg ET to 75 deg ET. With somewhat heavier clothing and a larger heat production it would be easily extended to the lower limits of the winter comfort zone. This can be seen from an experiment on nude subject J. D. H. With an increase of 36 per cent in his heat production, there would be no cooling of his nude body even at 67 deg ET (23 C). The comfort zone for men stripped to the waist and at rest has been shown

‡ Or 28 to 31 C.

[14] ASHVE RESEARCH REPORT No. 673—Determination of the Comfort Zone with Further Verification of Effective Temperature within this Zone, by F. C. Houghten and C. P. Yaglou. (ASHVE TRANSACTIONS, Vol. 29, 1923.)

[15] The Summer Comfort Zone: Climate and Clothing, by C. P. Yaglou and Philip Drinker. (ASHVE TRANSACTIONS, Vol. 35, 1929.)

[16] Heat Loss from the Body, by E. F. DuBois. (Harvey Lecture, Dec., 1938, *Bulletin* of the *New York Academy of Medicine*, Second Series, 15:143-173, 1939.)

[17] 21.2 C to 30.4 C, or 70.1 F to 86.9 F.

by C. P. Yaglou [18] to lie between 66 deg ET and 82 deg ET with an optimal value at 72.5 deg ET. His subjects were semi-nude and at a somewhat higher metabolic level (basal plus 36 per cent) than those of DuBois and Hardy. The findings agree well with the thought that the comfort zone as defined by the Society corresponds closely with the *neutral zone* of DuBois. The difference between the two is attributable largely to the level of heat production and the quantity and nature of the clothing.

The studies of Winslow and associates [19] have been made in a non-uniform environment and in one in which a source of radiant heat plays an important role. For these reasons it is somewhat more difficult to compare them to the experiments of the Society. Their subjects were at a *standard metabolism* somewhat higher than basal. These workers have been able to measure separately all partitions of heat elimination. They have found a zone of *thermal neutrality* 74 and 84 deg operative temperature for clothed subjects and 84 to 87.8 deg operative temperature for nude subjects. The partition of heat over the wide ranges of operative temperature studied are in essential agreement with those reported by DuBois and Hardy. The conclusions with reference to comfort are found in Professor Winslow's report of the Committee on Physiological Reactions, 1939 [20] and are quoted as follows:

> It should be expected that in the long run the comfort zone must coincide with the area of thermal adjustment with minimum physiological effort, and this adjustment, as noted, is reached for the clothed subject in a semi-reclining position, at approximately 80 F (26.7 C) operative temperature corresponding to 74 F (23.3 C) effective temperature (with 50 per cent relative humidity).

During the last two years, numerous experiments have been conducted on male nude subjects who were allowed to sleep in the experimental room over night. In this way they arrived at a steady state of thermal adjustments. The zones within which the temperature had to be maintained were 74 to 76 deg ET. A variation of half a degree above or below these limits evoked sensations of slight warmth or slight coolness.

The limits of the comfort zones as compared with the neutral zone of DuBois and zone of thermal neutrality are found in Table 2. From this table it is evident that the comfort zones as determined from sensations correspond closely to the zones of thermal neutrality as found by calorimetry.

It is quite a tribute to the research vision and technical ability of the group of investigators of the ASHVE that they were able to define a comfort zone based on subjective evidence that has remained an accepted standard and withstood criticism for a period of 17 years. It is of further interest that physiological and bio-physical experimentation did not establish a mechanistic basis for this zone until 12 years after its announcement. Although these studies are not complete, they do indicate that within the comfort zone a most important integration of physiological thermal adjustment occurs.

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[18] The Comfort Zone for Men at Rest and Stripped to the Waist, by C. P. Yaglou. (*The Journal of Industrial Hygiene*, 9:251-263, 1927.)

[19] The Influence of Clothing on the Physiological Reactions of the Human Body to Varying Environmental Temperatures, by A. P. Gagge, C.-E. A. Winslow and L. P. Herrington. (*The American Journal of Physiology*, 124:30-50, 1938.)

[20] Recent Advances in Physiological Knowledge and Their Bearing on Ventilating Practice, by C.-E. A. Winslow, T. Bedford, E. F. DuBois, R. W. Keeton, A. Missenard, R. R. Sayers and C. Tasker. (Report of ASHVE Research Technical Advisory Committee—ASHVE TRANSACTIONS, Vol. 45, 1939, p. 111.)

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TRANSPORT OF HEAT TO THE SURFACES

As previously stated, 75 per cent of the heat produced by the resting subject and probably 85 per cent by the working subject is developed within the muscles. These are located in the interior, so that the heat must be transported to the surface. The application of the physical laws to the transport of heat within the body was begun by Lefèvre [21] in 1911, continued by Burton in 1934,[22] Burton and Bazett in 1936,[23] Winslow and associates in 1937,[24] and Hardy [25] in 1937. Burton has noted that the flow of heat to the surface necessitates the existence of an internal *physiological gradient of temperature*. The flow takes place by the process of conduction across the tissues, and by convection by the blood flow from hotter to cooler parts. Through the studies of Bazett and McGlone [26] some knowledge has been obtained of the temperatures within the body. For the purposes of analysis, the body may be regarded as a cylinder (Hardy,[27]) with an interior throughout which the heat is generated and an external surface layer with a depth of 2 to 2.5 cm (0.8 to 1.0 in.). The temperature of the interior is, roughly, that of the rectum. It is not, however, always uniform. The addition of new heat in the muscles lying nearer to the surface may cause a local rise in temperature at these points. The surface layer consists of the epidermis beneath which lie the rich vascular beds of the dermis and subcutaneous tissues. Here the arteries break up into smaller arterioles and capillaries. From these zones, the cooled blood is collected in the venules and returned to the interior. Across the small distance of 2 to 2.5 cm with an environmental temperature of 28 C there is a drop of approximately 3.5 C (6.3 F) in temperature.

CONDUCTION OF HEAT

Burton [28] has noted that between the ranges of 20 to 30 C environmental temperature, there is little or no appreciable change in the resistance of the body to heat flow as shown by the constancy of his *circulation index*. Hardy [29] finds a constant value for the conductance of heat in a nude subject between an air temperature of 28 C and 22 C. Gagge and his associates [30] also find in the nude subject *thermal conductance* at a relatively constant value between 27 C T_0 and 20 C T_0. With a temperature difference of 4.5 C between the interior of the body and the external temperature of the surface, the conductance becomes constant and minimal. These findings all suggest that below an air temperature of 28 C and a skin temperature of 33 C, the transfer of heat

[21] Chaleur Animale et Bioenergetique, by J. Lefèvre. (Masson et Cie, Lib. de *L'Académie de Medécine*, Paris, 1911.)
[22] The Application of the Theory of Heat Flow to the Study of Energy Metabolism, by A. C. Burton. (*Journal of Nutrition*, 7:497-533, 1934.)
[23] A Study of the Average Temperatures of the Tissues, of the Exchanges of Heat and Vasomotor Responses in Man by Means of a Bath Calorimeter, by A. C. Burton and H. C. Bazett. (*The American Journal of Physiology*, 117:36-54, 1936.)
[24] Loc. Cit. Note 10.
[25] Loc. Cit. Note 6.
[26] Temperature Gradients in the Tissues of Man, by H. C. Bazett and B. McGlone. (*American Journal of Physiology*, 82:415-451, 1927.)
[27] The Physical Laws of Heat Loss from the Human Body, by J. D. Hardy. (*Proceedings of the National Academy of Sciences*, 23:631-637, 1937.)
[28] Loc. Cit. Note 22.
[29] Loc. Cit. Note 6.
[30] Loc. Cit. Note 19.

TABLE 2—COMPARISON OF COMFORT RANGES WITH ZONE OF THERMAL NEUTRALITY

INVESTIGATORS	EFFECTIVE TEMPERATURE		OPERATIVE TEMP.	REMARKS
	Optimum Line	Range	Range	
COMFORT ZONE				
Houghten and Yaglou	66	63–71	.....	Winter non-basal; at rest; normally clothed. Men and women
Yaglou and Drinker	71	66–75	.....	Summer non-basal; at rest and normally clothed. Men
Yaglou	72.5	66–82	.....	Entire year; non-basal; at rest and stripped to waist. Men
Keeton et al.	75	74–76	.....	Entire year; basal, nude. Steady state (9-hour exposure). Men and women
ZONE OF THERMAL NEUTRALITY—NEUTRAL ZONE				
DuBois and Hardy	75	73.2–76.9	.....	Basal; nude; men
	71.8	64.8–76.0	.....	Basal; clothed; men
Winslow, Herrington and Gagge	.....	.....	84.0–87.8	Non-basal; at rest; nude; men
	.....	.....	74 –84	Non-basal; at rest; clothed; men

across the surface zone can be explained satisfactorily by the physical laws governing conduction of heat.

The conductance of the peripheral tissues is defined by:

$$K = \frac{H}{A(T_R - T_S)}$$

K = Thermal conductance of tissue for depth of a surface zone of approximately 2.2 cm and for the character of tissue found in the average human subject.

H = Heat loss

A = Total surface area of body

T_R = Rectal temperature

T_S = Average skin temperature

The values for K in gram-calories per degree Centigrade per centimeter ^{2} per second as used by Lefèvre, Burton, Hardy and Winslow are respectively 0.00033, 0.0002, 0.00024, 0.0003 and agree closely with each other. As has been pointed out, these values are comparable to those for leather, paper, and cork. Hence, in the cold environments the surface zone of the body is a good insulator. From a physiological viewpoint, this must mean that a maximal vasoconstriction has occurred, and that we are dealing with bloodless tissues. The

tissues are, of course, not bloodless in the sense that they will not bleed when cut. However, the blood flow is so reduced that it does not convect any significant quantities of heat. Male subjects of normal body proportions therefore transfer all of their heat by conduction to the surface with the exception of the heat lost in vaporizing water from the lung surfaces.

It is next of importance to determine whether the changes in the character of the surface zone will influence the skin temperature or the conductance, when the subject is exposed to colder environments. The deposition of fat in the subcutaneous tissues would presumably decrease the tissue conductivity. Water or edema on the other hand should increase it. In the experiments of Winslow [31] the conductance values for the heavy subject (230 lb) showed approximately an average value of 11 kg-calories per square meter per hour. The lighter subject (105 lb) showed values of 13 kg-calories per meter [2] per hour for approximately the same skin temperatures. It is true that the heat production of the heavier subject per square meter was at a somewhat lower level. If, however, heat production is regulated by the necessity of keeping the body machine idling and in a state of preparedness for work at a given temperature level, one can find in this lower conductance value an explanation for the lower heat production. Hardy and DuBois [32] have recently studied three nude women over a range of 23 to 27 C environmental temperature. The heat production was the same as that of men over the same ranges. The skin temperature and conduction were lower. Women, as a rule, have more fat deposited in the subcutaneous tissues than men. The skin of the male rolls easily on the muscles. The women have a distinct padding beneath the skin. If a woman with a relatively fat surface zone and a man with a zone devoid of fatty deposits are producing heat at the same rate and are exposed to a cold environment, the female will retain more heat because of the greater resistance of her tissues to the flow of heat (lower conductance values). If she is to maintain a constant rectal temperature she will have to lower her skin temperature to a greater extent so as to produce a larger gradient. She should be able to tolerate a longer exposure to cold before she is forced to produce extera heat to maintain her thermal equilibrium. She would presumably have the lower range of her comfort zone extended below that of a man, just as the man's comfort zone could be extended by the addition of more clothes.

CONVECTION OF HEAT

Convection of heat is accomplished by the flow of warm blood into the surface zone. One may expect the increased blood flow to decrease the depth of the surface zone and to raise the skin temperature. Both of these effects will reduce conduction. Hence, in zones above 28 deg (neutral zone and the zone of evaporative regulation) conduction values will gradually decrease and be replaced by convection values. The relative magnitude of these values can be appreciated from an analysis of the blood flow of one of our subjects at rest in comfortable and various hot conditions.

It will be noted that the heat loss is divided into storage, conduction, and convection. The storage is figured in the usual manner from the mass, the rise in temperature and the specific heat. The difficulty arises in separating the

[31] Loc. Cit. Note 19.
[32] Loc. Cit. Note 7.

conduction and convection losses. The water content of the blood is somewhat higher than that of tissues. However, the osmotic pressures in the two are equal and therefore the conductivities are approximately the same. One can compute the heat loss by conduction through the superficial zone knowing (1) the interior temperature (T_R); (2) the surface temperature (T_S); (3) the conductivity of the superficial 2.2 cm layer of the body ($K = 0.00024$ gm-cal per degree Centigrade per centimeter^2 per second). The convection loss due to the blood flow through the surface zone may now be obtained by the difference between the total heat produced and the sum of the storage and conduction.

It will be noted (Table 3) that the conduction decreases by 70 per cent, the convection increases 138 per cent and storage appears as the environmental temperature is changed from a comfortable to a hot wet environment. The blood flow is increased by 5 to 8 times and the per cent of the cardiac output utilized for heat loss also is increased 5 times. As soon as there is significant storage the cardiac output and the pulse are increased. This means that the first adjustment to a hot condition is the increase in the quantity of blood used to convect the heat. When the environmental conditions become more difficult for heat loss, the rate of the circulation is increased. It may be assumed that the peripheral vessels have previously reached a state of maximum dilatation and that further heat loss can be effected only by increasing the rate of blood flow. Physiologists have noted that increase in work is associated with an increased cardiac output and an acceleration of the pulse. Physicians have long been accustomed to measure the smoothness with which the individual adjusts himself to a work level by taking the pulse before and after exercise tests of various types. In many studies made by the Society it has been noted that acceleration of the pulse occurred when the subjects were exposed to thermal stresses.

BLOOD VOLUME—SHIFTING OF BLOOD MASS

The potential capacity of the circulatory bed (arteries, veins and capillaries) is greater than the available blood. This is shown by the appearance of *shock* when the vascular bed is suddenly maximally dilated. The heart fails because it is unable to secure blood to maintain the circulation. Some of the blood channels must be regarded as quiescent reservoirs, which in periods of vasoconstriction contain little or no blood and in periods of vasodilatation contain larger quantities. The reserve capacity of these reservoirs can be further demonstrated by the intravenous injection into a normal subject of 2000 cubic centimeters of fluids without causing any embarrassment. When a subject is exposed to a hot environment some of the deep lying or visceral reservoirs are closed and the peripheral ones are opened. Thus the vital capacity of the lungs was shown [33] to be increased from 10 to 20 per cent on exposure to hot conditions. The quantity of blood in the lungs was less than in cold conditions. The counter part of this transfer of blood is shown by the not uncommon sensation of cold after a heavy meal. The blood has now been transferred to the viscera and the peripheral vessels have been constricted. The skin temperature falls and the individual becomes cold. These data indicate the extreme mobility of the circulating blood volume.

[33] Loc. Cit. Note 8.

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TABLE 3—CIRCULATORY ADJUSTMENTS TO HOT ENVIRONMENTS
(*Subject J. M.—Female*)

COND.	DRY-BULB F	WET-BULB F	HEAT PRODUCTION CAL/M²/HR	HEAT DISPOSAL CAL/M²/HR			BLOOD FLOW LITERS/HR	CARDIAC OUTPUT LITERS/HR	PER CENT CARDIAC OUTPUT FOR HEAT LOSS	CONDUCTANCE CAL/M²/HR/C	PULSE
				Storage	Conduction	Convection					
Comf.	84.0	64.6	36.3	0	24.5	11.8	4.2	144	2.9	12.8	63
Comf.	84.0	61.7	36.8	0	25.1	11.7	4.0	156	2.6	12.7	63
Hot Dry	99.4	65.0	37.3	3.3	9.5	24.5	22.3	168	13.3	30.9	70
Hot Dry	99.4	67.5	36.3	1.4	10.1	24.8	21.2	156	13.6	29.8	74
Hot Dry	99.5	71.6	36.7	1.6	10.4	24.7	20.2	174	11.7	28.8	68
Hot (no fever) Wet	98.7	84.5	35.3	3.4	7.2	24.7	29.7	162	18.3	38.4	74
Hot (fever) Wet	98.8	86.1	38.6	5.2	6.7	26.7	34.2	222	15.4	42.8	75
Hot (fever) Wet	99.0	87.2	37.0	5.2	7.7	24.1	27.1	201	13.5	35.7	74
Hot (fever) Wet	99.5	90.0	42.9	7.7	7.2	28.0	33.8	243	13.9	42.4	87

CHANGES IN BLOOD VOLUME

During the last year the changes in the blood volumes of subjects exposed to hot conditions have been studied by the dye injection method. The subject was placed in the room under comfortable conditions and a blood volume estimation was made. He was then exposed for varying lengths of time to hot conditions and the blood volume was redetermined.

It was found that a number of the subjects showed an increase (Table 4). This increase seemed to be due to blood coming from the reservoirs. The plasma, the proteins, and the mass of red cells added were quantitatively such as would be expected on this assumption. Apparently, there is a readily available quantity of blood in the storage depots which enters the circulation when the circulating volume must be expanded.

Under conditions favoring the evaporation of fluids a concentration of plasma volume was noted on a few occasions. However, in many experiments relatively larger quantities of water (2.5 lb) were lost through the skin without changing the blood volume. This return of the blood volume to its original value was apparently accomplished by a balancing of the water lost from the skin by that absorbed from the tissues. If the exposure to hot conditions is too prolonged then a concentration of the volume occurs such as is seen in subjects who suffer from heat stroke. One is impressed with the zeal with which the blood volume is protected. It is another one of the internal bodily constants the maintenance of which is indicative of a smoothly functioning machine.

DISPROPORTION BETWEEN BLOOD VOLUME AND CIRCULATORY BED

Last year Keeton and his associates [34] reported experiments which showed the development of a disproportion between blood volume and the circulatory bed induced by extreme degrees of capillary and venous dilatation. If a subject is placed in hot moist conditions where the ability to lose heat is taxed to the utmost, there is a maximal flow of blood through the surface zone. If this subject be suddenly changed from the horizontal to the upright position the veins are unable to contract adequately. The hydrostatic effect allows the blood to accumulate in the dependent extremities. The heart is deprived of blood, the rate is speeded up, and the arterial pressure falls. Simultaneously with this the blood flow through the brain is decreased, the subject becomes weak, dizzy and faints. This effect to a lesser extent is observed rather regularly in many individuals in the hot humid conditions of summer. Here the discrepancy is slight and the effects are chronic. Such subjects may present themselves to a physician for treatment because of fatigue, lassitude and dizziness. This type of discrepancy due to a lack of venous tone, which we have advanced as the explanation for heat exhaustion, is to be sharply distinguished from a discrepancy due to excessive loss of fluids by sweating which is the forerunner of heat stroke.

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[34] ASHVE RESEARCH REPORT No. 1151—The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion, by R. W. Keeton, F. K. Hick, Nathaniel Glickman and M. M. Montgomery. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

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TABLE 4—PLASMA VOLUME INCREASED

SUBJECT AND DATE	CONDITIONS		EXPOSURE TIME IN MINUTES	PLASMA VOL. (CC)	PER CENT CHANGE	RBC VOL. (CC)	PER CENT CHANGE	RECTAL TEMP. F	WEIGHT LOSS IN GRAMS
	Dry-Bulb F	Wet-Bulb F							
C. G.	Comf.[a]		...	3049	...	2485	...	98.3	...
6-3	99.0	71.7	130	3448	+13.1	2731	+ 9.9	99.4	——
A. M.	Comf.		...	2583	...	2349	...	98.9	...
9-23	99.6	81.5	59	2866	+10.9	2742	+16.7	99.0	381
F. C.	Comf.		...	2840	...	2232	...	97.8	...
9-5	99.2	75.5	162	3122	+ 9.9	2589	+16.0	98.5	596
H. C.	Comf.		...	3367	...	2468	...	98.4	...
6-7	98.9	75.7	138	3663	+ 8.8	2855	+15.7	99.4	——
F. C.	Comf.		...	3380	...	2575	...	97.4	...
7-26	112.5	79.5	216	3572	+ 5.7	2832	+10.0	98.4	——
H. C.	Comf.		...	3012	...	2802	...	98.6	...
5-28	99.0	68.2	160	3165	+ 5.4	2924	+ 4.4	99.5	——

[a] Comf. = Dry-bulb 83.5, wet-bulb 67.5

DANGERS ARISING FROM CHARACTER OF THE ENVIRONMENT

Up to this point, the discussion has been directed towards studies attempting to solve the problems of comfort. It is important to briefly consider the manner in which the environment may bring harm to an individual. It is obvious that the potential dangers lie in the hot zones. The vulnerable link will be found in the convection transport of the heat by the circulation. The experiment on work by Hardy and DuBois [35] in a relatively cool environment (68.9 F) showed that there is a large storage of heat when the work is violent. If the environmental conditions are more difficult, less violent work will cause storage. Fleisher, Stacey, Houghten, and Ferderber [36] found that the most significant indices of danger for workers in high effective temperatures were a rise in body temperature (storage) and an increase in pulse rate. If one reverts to the previous discussion of the convection of heat by the circulation, it can be seen that a significant rise in pulse may indicate an increase in cardiac output. If this rise is excessive it means that the output has probably reached its limit. The rise in temperature, of course, is indicative of heat storage. It is obvious that this cannot be continued indefinitely.

These investigators report a drop in the vital capacity in the hotter atmospheres amounting to maximum of 8 per cent. The changes were not regarded as significant. It should be noted that when the heart becomes inefficient and fails to pump forward the blood received from the lungs, the vital capacity falls. Clinicians regularly use this as a test for cardiac function. Hence, it

[35] The Effect of Exercise and Chills on Heat Loss from the Nude Body, by J. D. Hardy, A. T. Milharat and E. F. DuBois. (*Journal of Nutrition*, 16:477-492, 1938.)

[36] ASHVE RESEARCH REPORT No. 1106—Air Conditioning in Industry—Physiological Reactions of Individual Workers to High Effective Temperatures, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

might be worth while studying the vital capacity under the above conditions more closely as a possible index of heart failure and therefore as a danger sign.

The effects of exposure of an individual at rest to high effective temperatures have been studied by McConnell and Houghten [37] and by McConnell, Houghten and Phillips.[38] They attempted to find in changes in the physiological reactions (rectal temperatures, pulse rate, blood pressure) signs of danger to the subject.

The question is often raised as to the danger of entering and leaving air conditioned buildings in the hot humid summer weather. The effect has been described by some writers as *shock-like* in character. It is apparent from the studies which have been reported by the authors that an analysis has been made of the circulatory adjustments as one moves from a steady state in a comfortable environment to a steady state in hot conditions. This analysis is not complete, and will require much more work. When these adjustments are known, it will then be easier to follow the more rapid changes required by the sudden emergence from an air conditioned space on a hot summer day. New experimental methods will doubtless have to be developed to answer this question but the information should be of value to engineers.

THE PROBLEM OF COMFORT

It would be very helpful to have some method of measuring objectively the sensation of comfort. In a previous section of the paper it was shown that the comfort zones agree well with the zone of thermal neutrality which has also been designated the zone of vaso motor control. When the subject, either nude or clothed, passes to the cold side of this zone, a sensation of coolness appears. If the coolness is replaced by a sensation of cold, then the subject recognizes the state as one of discomfort. One should therefore realize that comfort or discomfort is not a sensation, but an affective state applied in this instance to a sensation of cold. In the same manner a given taste in the mouth may be recognized as sour or sweet. In the next breath, the individual describes it as pleasant or unpleasant. Pleasant is not a sensation but is a description of the effect produced on the subject by the taste sensation. Most of the investigators agree and the experiments of Winslow show quite clearly that on the cold border of the comfort zone the skin temperature and presumably the stimulation of the cold spots give rise to the feeling of discomfort.

When one approaches the warm end of the comfort zone the answer to the question is not so obvious. Thus Winslow found an excellent correlation in the clothed subject between the skin temperature and the sensation of *thermal pleasantness*. In the nude subject the correlation with the skin temperature was also good, but it was better with *degree of wettedness*. Here, there are apparently two physiological processes, a sensation of warmth produced by a stimulation of warm-spots, and the secretion of sweat, to which the sensation of unpleasantness can be attributed. Perhaps it will help to say again that discomfort is an affective state and that it may be associated with either of these processes or some other physiological adjustment that occurs simultane-

[37] ASHVE RESEARCH REPORT No. 654—Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten. (ASHVE TRANSACTIONS, Vol. 29 1923.)

[38] ASHVE RESEARCH REPORT No. 672—Further Study of Physiological Reactions, by W. J. McConnell, F. C. Houghten and F. M. Phillips. (ASHVE TRANSACTIONS, Vol. 29, 1923.)

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ously. From a practical viewpoint, there are many advantages in correlating discomfort with the appearance of sweat or with the degree of wettedness. Anatomically the cold spots (Krause's end bulbs) for the reception of cold sensations are located very close to the surface. The warm spots (Ruffini's end organs) lie deeper in the skin or subcutaneous tissue. They have been shown by Bazett, McGlone, Williams and Lufkin [39] to be the receptors for sensations of warmth. The cold spots are admirably located for stimulation by the environmental air. The warm spots on the other hand are located in the area where marked fluctuations in blood flow occur.

It will now be helpful to consider the problem of why many people are not comfortable in the comfort zone.

The heat production of the individual may be below normal. This can be due to a lack of secretion of the thyroid gland. Such people are notoriously cold in the customary environmental temperatures. They keep the windows closed, wear stockings in bed and fur coats in mild weather. If an individual has eaten poorly, lost weight and become emaciated, his heat production falls to conserve energy. He too suffers from cold. In this group fall elderly people and certain other individuals with unusually active sympathetic nervous systems. All of these will enjoy a warmer environment. They will have their comfort zone moved upward into the warmer temperature ranges.

In any large group of people there are a number who are unusually active. They move around more than usual. Their heat production is increased by voluntary activity. Other individuals will be found who eat heavily and have increased heat productions, due to the food. Finally, there are others in whom the heat production may be elevated because of over-activity of the thyroid gland (goitre). Such individuals are unusually warm in the average comfort zone. For them, comfort is to be found over lower temperature ranges. Women who have more fat in their subcutaneous tissues than men have wider than normal comfort zones with extensions on both ends. Finally, fat men have their comfort zone moved downward into colder environments.

One can now easily appreciate the view that the comfort zone is not one with fixed rigid borders, but a phasic zone with extensions and recessions on either end. In closing, I would like to quote from Bedford: [40]

> The feeling of warmth experienced by a person is not dependent solely on the warmth of his environment: a variety of other factors also affect his state of comfort. Differences in clothing, in muscular activity, in nutrition, age, and general bodily build, and, not least, in acclimatization, all tend to influence the feelings of warmth experienced by different persons in the same environment; and in consequence it often happens that in a room in which most people are comfortable some feel too warm while others are too cool.

[39] Sensation, by H. C. Bazett, B. McGlone, R. G. Williams and H. M. Lufkin. (*Archives of Neurology and Psychiatry*, 27:489-517, 1932.)

[40] The Warmth Factor in Comfort at Work, by Thomas Bedford. (Report No. 76, H. M. Stationery Office, London, 1936.)

No. 1176

COMFORT AND HEALTH AND TEMPERATURE A MATHEMATICAL SOLUTION

By W. L. FLEISHER* AND W. L. FLEISHER, JR.**
NEW YORK, N. Y.

COMFORT and temperature have been so closely associated from time immemorial, that there seems no escape from the fact that there is a close relationship between them. Humidity, too, has always been correlated with this same temperature and undoubtedly plays a major part in the feeling of comfort.

The adjustment of man to his environment, the maintenance of an equilibrium temperature, his departure from a normal temperature in times of sickness and disease, his efficiency, his irritability in response to the weather or at times of storm, all point to a fundamental relationship between man and his environment. It is a fascinating problem and the purpose of this paper is to try to offer a solution.

As temperature is the basis of this discussion it is proper to start with a definition of temperature as related to heat as recognized by the physicists. They define the thermodynamics of temperature, as abbreviated from Haas' *Introduction to Theoretical Physics,* as follows:

"In every case in which energy is distributed among a very large number of individuals in a manner not directly perceptible to human senses, this energy may be interpreted as heat. This heat energy, when distributed throughout bodies, is identified with molecules. All quantities which, in a statistical sense, are proportional to the number of individuals must, therefore, in a thermal sense, be proportional to the number of molecules or to the mass. From the electron theory that the forces acting between the particles of matter are central forces depending only upon the distance between the particles, a system of this kind is governed by the Principle of the Conservation of Energy. If the internal forces only depend on the distances separating the individual molecules, then the internal energy will evidently depend only on one parameter, viz., on the volume of the unit of mass, or the so-called specific volume."

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** Research Engineer, Air & Refrigeration Corp.

Presented at the 47th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., January 1941.

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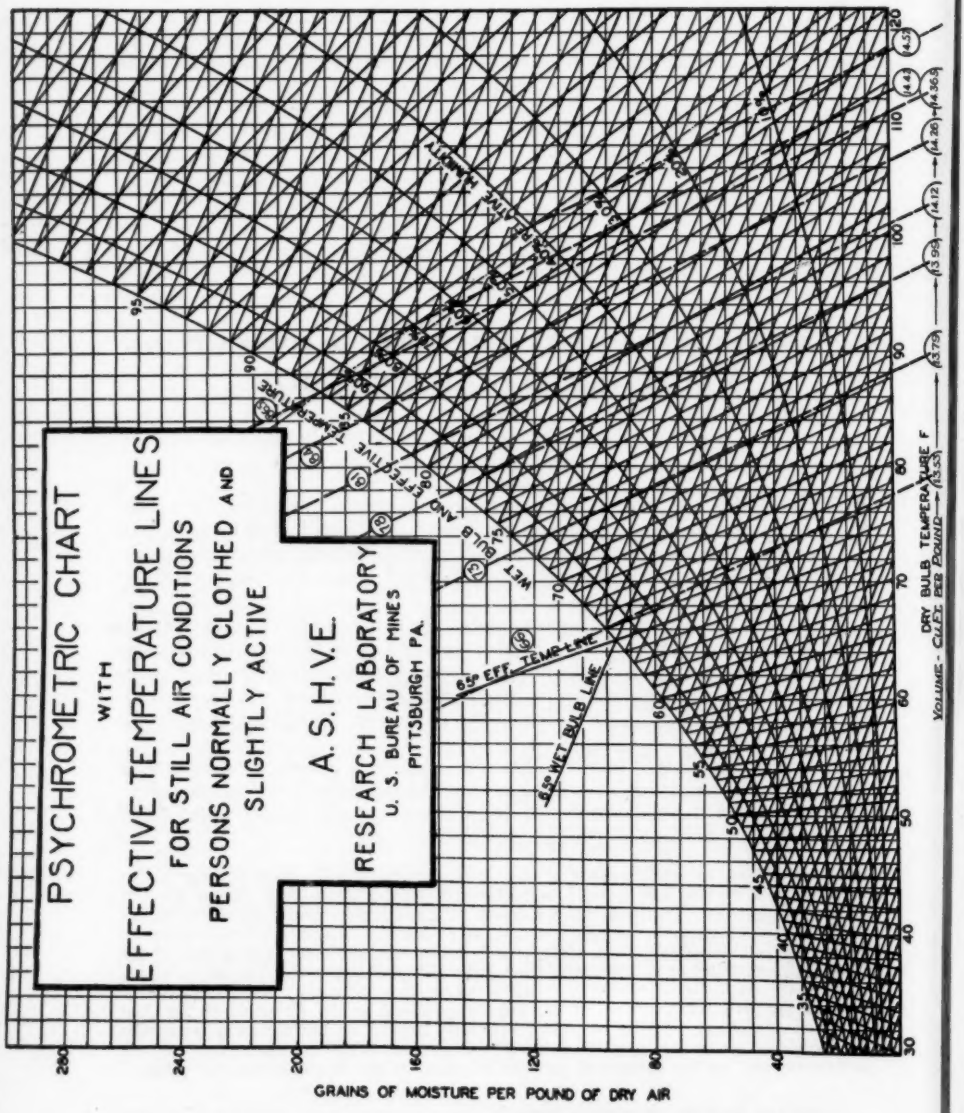

FIG. 1. PSYCHROMETRIC CHART WITH EFFECTIVE TEMPERATURE AND CONSTANT VOLUME LINES

Haas goes on to show that in a perfect gas, if a system of molecules is considered, the energy of which may be regarded as purely kinetic, the theorem of equipartition leads to the two relations

$$E = \frac{s}{2} NkT \quad \ldots\ldots (1)$$

and

$$pV = NkT \quad \ldots\ldots (2)$$

In these equations N denotes the number of molecules, k Boltzmann's constant, and s the number of degrees of freedom of the molecules. From Equation (2) an important conclusion can be derived, to the effect that at equal pressure and equal temperature equal volumes of gases must contain the same number of molecules, and that at a given pressure and given temperature the densities of different gases are in the ratios of their molecular weights.

In a further development of this formula, it will be noted that

$$pV = NRT \quad \ldots\ldots (3)$$

Since the energy of a homogeneous body must, in general, be determined by volume and temperature, it follows that the derivative of

$$\left(\frac{\delta E}{\delta V}\right) = 0.$$

From this formula can be calculated approximately the dependence of the vapor pressure on the temperature and that degrees of temperature represent intensities, not quantities, of heat.

As previously indicated, this paper will deal with two rather abstract qualities, comfort and temperature, and as temperature translated from its heat source seems to affect individuals through the surrounding air, it is logical to analyze the properties of an air-water vapor mixture from a thermodynamic angle.

Before beginning this analysis it is desirable to emphasize the reason for the investigation by relating the theory to the comfort and health of human beings. In studying what conditions of the atmosphere or what ambient conditions in enclosures produced comfort or discomfort, physiological changes or dislocations, the authors studied the literature on the subject found in the TRANSACTIONS of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, dating back over a period of 20 years. Starting with the reactions of human beings to high temperatures and high humidities, Society researchers turned to normal temperatures and varying humidities. The TRANSACTIONS of the Society from 1923 to 1939 have numerous technical papers each year on this research problem. In 1938 and 1939 the Committee on Research of the Society renewed its investigations on the reactions of workers in industry to imposed conditions of the surrounding air.

Out of this research was developed a zone of comfort for winter and summer and a series of effective temperature (ET) lines. This work represents an outstanding contribution of the Society to scientific research.

An explanation of the effective temperature lines is essential to the understanding of the development outlined herewith. THE GUIDE 1940 of the Society defines effective temperature: ". . . an arbitrary index which combines into a single value the effect of temperature, humidity and movement of air on the degree of warmth or cold felt by the human body." To be more specific, a psychrometric chart is shown in Fig. 1 giving effective temperature lines for persons normally clothed and slightly active, as developed over many years by the ASHVE Research Laboratory, Pittsburgh, Pa. Design conditions for most cooling installations in the summertime indicate that a 73 deg ET, which varies over a wide range of relative humidities, is the most satisfactory for most

TABLE 1—TABULATION OF PSYCHROMETRIC CONDITIONS CORRESPONDING TO VARIOUS EFFECTIVE TEMPERATURES

Effective Temp. Deg ET	Relative Humidity Per Cent	Temperature Deg Fahr			Grains per Pound	Constant Volume Cu Ft-Lb (Dry Air Plus Water Vapor)	Constant Volume Corres. to ET at Saturation
		Dry-Bulb	Wet-Bulb	Dew-Point			
73	70	76.1	68.9	65.6	95	13.79	13.79
	50	78.8	65.7	58.6	74	13.79	13.79
	30	81.7	61.7	47.5	49	13.79	13.79
78	70	82.1	74.3	71.1	116	14.01	13.99
	50	85.6	71.1	65.0	94	14.03	13.99
	30	89.6	66.8	54.0	62	14.04	13.99
86.5	70	92.2	83.8	81.1	163	14.42	14.365
	50	97.2	81.0	75.5	134	14.47	14.365
	30	103.8	77.3	66.0	96	14.52	14.365

people. Given in Table 1 are the temperature and moisture conditions for extreme points of relative humidity on the 73 deg ET line—that is, high and low, which show that at 70 per cent relative humidity there is a dry-bulb temperature of 76.1 F, a wet-bulb temperature of 68.9 F, a dew-point temperature of 65.6 F and 95 grains of moisture per pound of dry air. At 50 per cent on the same line, there is a dry-bulb of 78.8 F, a wet-bulb of 65.7 F, a dew-point of 58.6 F and 74 grains of moisture per pound of dry air. At 30 per cent on the same line, there is a dry-bulb of 81.7 F, a wet-bulb of 61.7 F, a dew-point of 47.5 F and 49 grains of moisture per pound of dry air. If total heat, as represented by the wet-bulb, is the measure of comfort, how is it that on this effective temperature line variations exist from 68.9 F wet-bulb to 61.7 F wet-bulb, or total heats of 33.9 Btu per pound and 27.5 Btu per pound of dry air with vapor to saturate it.

In further investigations [1] of the reactions of human beings, the Research

[1] ASHVE Research Report No. 1106—Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber. (ASHVE TRANSACTIONS, Vol. 45, 1939.)

Laboratory of the Society and its co-operating groups discovered that at 86.5 deg ET a marked physiological change took place in all of the persons subjected to this condition. An examination of the variables on this effective temperature line show in Table 1 that at 70 per cent relative humidity, the dry-bulb is 92.2 F, the wet-bulb is 83.8 F, the dew-point 81.1 F and 163 grains of moisture per pound of dry air. At 50 per cent relative humidity the dry-bulb is 97.2 F, the wet-bulb 81 F, the dew-point 75.5 F

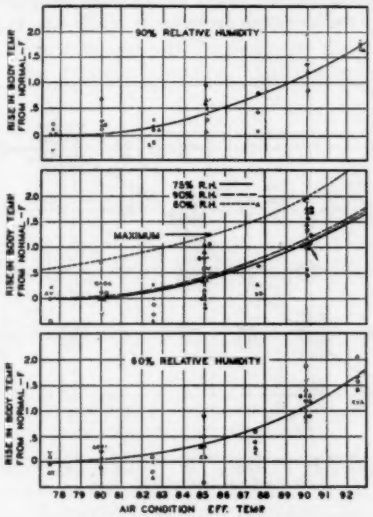

FIG. 2. RELATION BETWEEN RISE IN BODY TEMPERATURE FROM NORMAL DURING THREE HOURS EXPOSURE AND EFFECTIVE TEMPERATURE OF THE AIR CONDITION IN TEST ROOM FOR 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES. TEST POINTS FOR INDIVIDUAL SUBJECTS APPLY TO HEAVY SOLID CURVES

and 134 grains of moisture per pound of dry air. At 30 per cent relative humidity the dry-bulb is 103.8 F, the wet-bulb 77.3 F, the dew-point 66 F and 96 grains of moisture per pound of dry air and a corresponding difference in the total heats. It would seem that there was no relationship whatsoever in these different points, but how to account for the fact that human beings have reactions almost exactly alike as to comfort, when comfortable, or physiological changes when such changes take place at each of these combinations, is a subject worthy of investigation, if the many tests that had been made on human reactions or physiological changes were sound. The common factor of comfort at these different conditions or the common factor of marked physiological change and discomfort at such seemingly dissimilar conditions

were too evident to be merely experimental, empirical or simply chance. What was there at these various points on an effective temperature line that was similar and that would give a clue to the markedly similar reactions of human beings at these diverse conditions? Strangely enough and strange only because it appears that it has not previously been noted, all points on an effective temperature line from 100 per cent relative humidity to zero per cent relative humidity *are on a constant volume line* (volume of 1 lb of air plus the volume of the water vapor).

Referring to the psychrometric chart of effective temperatures in Fig. 1 for lines of 73 deg ET, the constant volume is 13.79 cu ft per pound for all

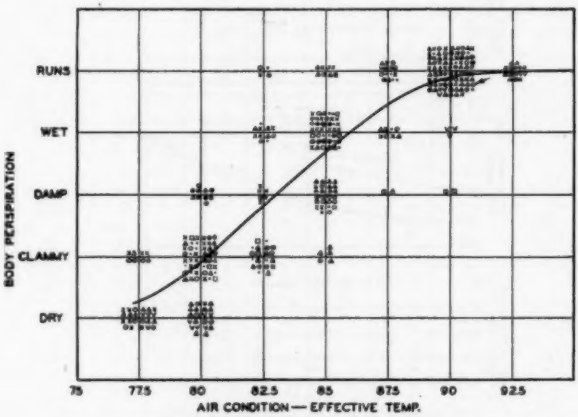

FIG. 3. RELATION BETWEEN DEGREE OF PERSPIRATION ON BODY AFTER THREE HOURS EXPOSURE AND EFFECTIVE TEMPERATURE OF AIR IN TEST ROOM. FOR ALL SUBJECTS IN TESTS AT 60, 75 AND 90 PER CENT RELATIVE HUMIDITY

points from 0 per cent relative humidity to 100 per cent relative humidity. This same relationship holds true for any other effective temperature line. In other words, as long as the absolute pressure remains constant people react exactly the same to the same constant volume (by constant volume is meant the cubic feet occupied by a pound of dry air and its moisture or water vapor).

Where the effective temperature line deviates from a constant volume line it is felt that the effective temperature line is in error because the evidence is all in favor of the theory that these points all lie on a constant volume line.

Next consider the 86.5 deg ET line where a strange reaction takes place physiologically. The theory shows that this effective temperature line lies or should lie on a constant volume line of 14.365. Actually, the effective temperature line as developed by the Society varies from 14.42 at 70 per cent relative humidity to 14.52 at 30 per cent relative humidity. As the physio-

logical tests at this line were indicated as uncomfortable by all the subjects, there was only one way to develop this line empirically, and that was by exactly similar physiological reactions. If new tests are conducted on this theory, it is believed that a much closer relationship to a constant volume line would be found to exist. It is apparent that a definite physiological change had taken place in every one of the individuals subjected to these conditions.

Next consider a critical effective temperature line, that is, the 78 deg ET line. Here, according to the records of recent tests,[2] physiological changes started which correspond to a constant volume of 13.99 cu ft. This compares with a dry-bulb temperature of 78 F at 100 per cent relative humidity and 96 F at 0 per cent relative humidity. Special attention should be given

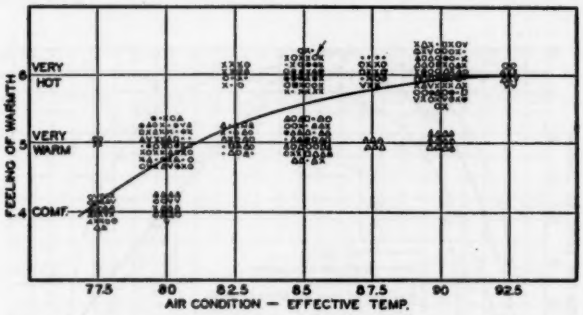

FIG. 4. RELATION BETWEEN FEELING OF WARMTH AND EFFECTIVE TEMPERATURE OF AIR IN TEST ROOM FOR ALL SUBJECTS AT 60, 75 AND 90 PER CENT RELATIVE HUMIDITY

to the dry-bulb temperature at 0 per cent relative humidity. As soon as the constant volume line at 0 per cent relative humidity corresponds to the blood temperature, the entire physiological reaction of the body changes. It probably is easier to determine when a change takes place by projecting the constant volume line to the 0 per cent relative humidity line than by any physical means. Above the 78 deg ET line, or rather the 13.99 constant volume line, it will be found that the body no longer can establish a simple balance, and other factors are introduced to compensate. Charts from previous tests[3] shown in Fig. 2, 3 and 4 indicate that at 77.5 deg ET a rise in body temperature, a feeling of warmth and body perspiration take place, all indicating the *beginning* of a physiological change.

By taking the statistical results of field tests[4] made over a two-year period with 274 subjects, it will be noted in Fig. 5 that over 75 per cent of all

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[2, 3] Loc. Cit. Note 1.

[4] ASHVE Research Report No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley (ASHVE TRANSACTIONS, Vol. 44, 1938.)

subjects are comfortable at about 70 deg ET and that although about 40 per cent are comfortable at 75 deg ET, none are comfortable over 77.5 deg ET.

The constant volume theory is not to be construed as the underlying disclosure of this paper indicating expected reactions from an effective temperature scale. However, the closeness of the location of the constant volume lines to the effective temperature lines was the clue to the basic discovery.

The theory of the kinetic energy of gases is the basis of the mathematical theory developed in this paper. Heat is a form of energy—or probably the kinetic energy of confused oscillating motions of the particles of matter. It is necessary to determine the part constant volume contributes in the energy formula for gases.

It has already been indicated that the pressure exerted by any perfect gas is a function of the number of molecules per unit of volume, the mass of the

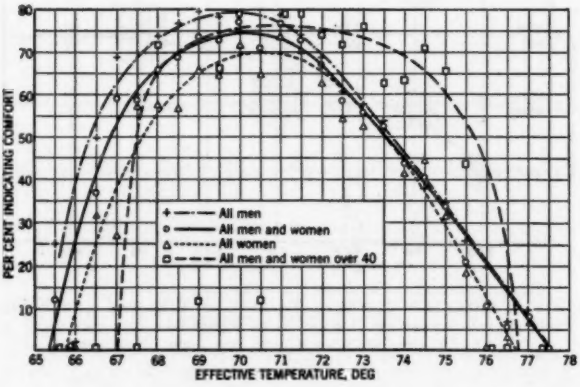

FIG. 5. RELATION BETWEEN EFFECTIVE TEMPERATURE MAINTAINED IN CONDITIONED SPACE AND PERCENTAGE OF OBSERVATIONS INDICATING COMFORT FOR PERIOD AUGUST 2 TO SEPTEMBER 3, 1937. CURVE FOR ALL MEN AND WOMEN UNDER 40 COINCIDES WITH CURVE FOR ALL MEN AND WOMEN

molecules of the particular gas considered and the mean of the square of the velocity of the molecule. This can be developed, as very simply expressed by Hausmann and Slack in their *Kinetic Theory of Gases* into the formulae which follow.

It is known that energy is needed to raise the temperature of a substance, and thus the temperature of a gas is assumed to be directly proportional to the kinetic energy of the gas molecules. The intermediate velocity v must, therefore, be such as to impart the same kinetic energy to N molecules, each of mass m, as that due to their individual velocities, v_1, v_2, . . . v_N

Hence,

$$N\,(\tfrac{1}{2}mv^2) = \tfrac{1}{2}mv_1^2 + \tfrac{1}{2}mv_2^2 + \ldots \tfrac{1}{2}mv_N^2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

The varying individual velocities may therefore be replaced by a single velocity which is found by squaring the individual velocities, taking the mean of these squares, and then extracting the square root. The result is known as the root-mean-square or *rms* velocity.

Consider a cubical box of volume $V = l^3$, containing N molecules of gas each of mass m, and let the rms velocity of the gas molecules be v. The actual motions of the molecules within the box can be resolved into three components, as though one-third of the molecules were moving perpendicularly between each pair of opposite faces. Such a group of molecules moving between face A and that opposite would encounter many collisions on the way. Suppose the collisions to occur after traveling a small uniform distance x from A. The number of molecules in a zone of this thickness would be $\frac{x}{l} N$, one-third of which would strike A with a velocity v and rebound with a velocity of $-v$. Each of these molecules thus undergoes a change of momentum of $2\ m\ v$, which is accordingly equal to the impulse imparted to the wall by collision. Including all of the molecules striking face A, the total impulse in time $t = \frac{2x}{v}$ is

$$Ft = \tfrac{1}{3} \times \frac{x}{l} N \times 2mv \qquad (5)$$

The molecules travel $2x$ in a time $\frac{2x}{v}$ and strike $\frac{v}{2x}$ times per second. Total impulse per second is therefore = to F, and

$$F = \frac{xN}{3l} \times 2mv \times \frac{v}{2x}$$

by factoring, therefore

$$F = \frac{Nmv^2}{3l}$$

and the pressure on the face

$$= P = \frac{F}{l^2} = \frac{Nmv^2}{3l^3} = \frac{Nmv^2}{3V}$$

or

$$PV = \frac{Nmv^2}{3} \qquad (6)$$

The kinetic theory of gases can be expanded for a mixture of gases such as air and water vapor by the use of Dalton's law of partial pressures.

$$P = P_1 + P_2 + P_3, \text{ etc.} \qquad (7)$$

or the total pressure equals the sum of the partial pressures in a gas mixture.

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The pressure of the air surrounding human beings $= P_1 = \frac{1}{3}(N_1 m_1 v^2)$ and the pressure of the water vapor is

$$P_2 = \frac{1}{3}(N_2 m_2 v_2^2)$$

Therefore, substituting in Equation (7) the values of P_1, P_2, etc.

$$P = \frac{1}{3}(N_1 m_1 v_1^2 + N_2 m_2 v_2^2, \text{ etc.}) \quad (8)$$

and

$$P\frac{V}{w} = \frac{1}{3}\frac{V}{w}(N_1 m_1 v_1^2 + N_2 m_2 v_2^2) \quad (9)$$

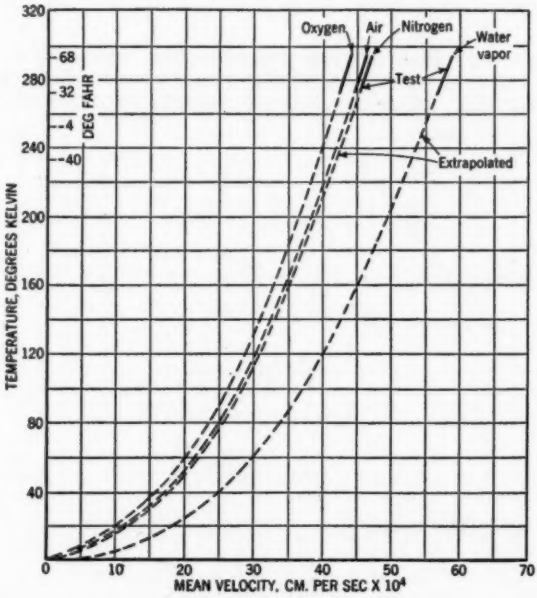

FIG. 6. MEAN VELOCITY CURVES FOR KINETIC ENERGY EQUATION FOR AIR AND ITS COMPONENTS

This gives the specific energy of the gas mixture which is bombarding the human body. At constant barometric pressure P and constant specific volume $\frac{V}{w}$ the total specific kinetic energy is constant, regardless of the molecular composition of the mixture.

Equation (9) does not give the complete analysis. While it interprets physiological reactions in terms of basic kinetic energy, it does not explain it in terms of temperature. Curves shown in Fig. 6 give the relation between the mean

velocity of the various component gases and the sensible temperature. It will be noted that for any temperature t the quantity $m\ v^2 =$ a constant. From Fig. 6 at 68 F air has a molecular weight of 28.97 and at this temperature 68 F a mean velocity of 4.62 $cm \times 10^4$ per second. At this temperature $m\ v^2$ of air equals $m_1 v_1^2$ of water vapor at the same temperature,[5] or $H_2O = 18 \times (5.87)^2$ and air should equal $28.97 \times (4.62)^2$. Actually the readings are slightly incorrect, but are close enough to illustrate the theory. For any air-water vapor mixture, the air and water vapor are at the same temperature so that all the molecules in the mixture have the same kinetic energy. This means that any mixture, from the purely kinetic energy viewpoint, can be considered as a single gas, and the equivalent temperature point is at 0 per cent relative humidity. Therefore since the body reacts to the atmosphere purely on an energy basis, the equivalent temperature at 0 per cent relative humidity is the temperature or energy to which the body responds. This can be shown mathematically:

$$v = f(t) \text{ (when } v \text{ is the mean velocity of the molecule) (See Fig. 6)} \quad (10)$$

and

$$m_1 v_1^2 = m_2 v_2^2 = C \text{ at the same temperature} \quad (11)$$

If Equation (9) is solved in terms of Equation (11),

$$P \frac{V}{w} = \frac{1}{3} \frac{V}{w} m_1 v_1^2 (N_1 + N_2) \quad (12)$$

and substituting Equation (10) in Equation (12),

$$P \frac{V}{w} = \frac{1}{3} \frac{V}{w} m_1 [f(t)]^2 (N_1 + N_2) \quad (13)$$

Equation (13) places the energy equation in terms of a single gas and its corresponding temperature, that is, at 0 per cent relative humidity where no water vapor exists.

An infinite number of relationships can be developed between air and water vapor and the energy of bombardment per unit volume, where constant volume and pressure remain constant, will still have the same specific kinetic energy. In other words, the moisture content in a pound of air can vary from 0 to 100 per cent saturation without changing its kinetic energy bombardment, as long as the barometric pressure and the constant volume remain constant. The effect of an air and water vapor mixture under constant barometric pressure can be foretold by determining the constant volume of the mixture. It will be seen from this deduction, for instance, that with lower barometric pressure greater constant volume will be required to balance the equation or at great heights more heat is required to obtain the same energy. In deep mines, lower constant volumes are required for the same balance.

It was previously mentioned that the 78 deg ET line, at which line physiological changes in the human body begin, crosses the 0 per cent relative humidity line at about 96 F. The kinetic energy given out by the body corresponds to an energy represented by a heat level of 96 to 98 F. The protective level of the body against its environment is therefore positive up to

[5] The values of 5.87 and 4.62 are constants of the kinetic theory of gases.

an energy level from without, represented by the mathematical equivalent of the energy of the air and water vapor part of the equation under constant barometric pressure of 29.92 in. of mercury corresponding to the numerical value of 13.99 cu ft per pound of dry air and its water vapor. When the kinetic energy exceeds that equivalent, the temperature of the body rises to ward off the introduction of too much energy or the absorption of too many energizing units (corpuscular theory of bombardment). The determination of the wave lengths creating these conditions is under consideration for further investigation.

If this theory is applied to the reactions of human beings and the limiting conditions of comfort are taken from the ASHVE comfort chart, *no one* at rest in winter feels comfortable below 66 deg ET. This from Equation (13) occurs at a temperature of 77.5 F and 0 per cent relative humidity, and no one under the same conditions of rest feels comfortable over 79 deg ET, almost exactly equivalent to 98 deg at 0 per cent relative humidity (the reason for the 77.5 F equivalent relationship is not brought out in this paper).

The theory is further confirmed by the fact that premature babies require an 85 F dry-bulb temperature and 65 per cent condition in order to complete their development. It will be noted from the mathematical development in this paper that this corresponds to a bombardment equivalent to 98.6 F on the 0 per cent relative humidity line.

There may be some criticism of the basic theory in that at higher air movements, the adjustment to imposed conditions appears to be at a higher effective temperature. This can be explained by the theory that the primary adjustment is through a vapor film and an air film, and that when a high wiping effect is achieved by the movement of a moving film over a stationary film, the molecular pattern is changed and a greater number of heat or energy waves can penetrate from the higher to the lower environment.

The theory has its greatest value in that it orients feelings of comfort, or physical reactions to the surrounding air, in terms of physical laws and a mathematical equation relating comfort to the physical matter or molecules which constitute the atmosphere. Effective temperatures have been purely empirical up to this time. The kinetic energy theory definitely establishes the validity of the observations along the effective temperature lines and beyond everything else establishes a method of investigating the physiological processes involved in the life processes.

DISCUSSION

J. H. CARTER (WRITTEN): This paper is potentially of fundamental importance, and it deserves searching and rigorous examination. The authors can assist by revising their paper, making it more complete and comprehensible.

My main criticism is that the paper as presented is not clear. There are, for example, several references to *the theory*, but there is no statement which in itself tells us just what the theory is. We have to deduce what it is that the authors are trying to prove. Likewise there is no conclusion, that is, there is no summing up of what has been proved or presented.

There are lacunae in the mathematical development. As an added irritation, there are practically no identifications of symbols used. I do not contend that a subject

of this sort should be written down to the lay mind, but it certainly ought to be presented so as to be reasonably comprehensible to the specialist. It happens that this paper was the subject of earnest study and discussion by several well-informed members of the St. Louis Chapter, as well as a research physicist who has specialized in this field, and the unanimous report was that they could not follow the train of thought. We all have the feeling that the authors know what they are about, and that they have something here, but there are just too many gaps for us to jump.

A. R. BEHNKE[6] (WRITTEN): The discovery of a simple physical basis underlying physiological reactions associated with health and comfort would of course constitute a fundamental contribution both in engineering and medicine.

Thus far, however, no simple physical expression has been found which can be used to predict comfortable or even harmful air conditions. Engineers have been forced to depend upon physiological responses as the basis for ventilation and air conditioning. Even the effective temperature concept corrected for radiant heat does not apply in engine or fire rooms where spot cooling plays an important role.

TABLE A

Effective Temperature	Relative Humidity	Dry-bulb Temperature	Specific Volume Dry Air[a]	Specific Volume Moist Air[b]	Fleisher Constant Volume[c]	Total Weight, Lbs of Mixture in V_{as}	Fractional Weight of Water[d]	Molar Fraction of Water[e]	Humidity Ratio ASHVE Guide, 1942
Deg. ET	Per Cent	Deg. F	V_a	V_m	V_{as}	W_{as}	f_w	A_w	W
73	100	73	13.42	13.55	13.79	1.01748	0.01718	0.0274	0.0176
73	70	76.3	13.50	13.60	13.79	1.0137	0.0135	0.0216	0.0138
73	50	78.8	13.56	13.65	13.80	1.01067	0.01056	0.0169	0.0107
73	30	81.7	13.63	13.70	13.80	1.00730	0.00725	0.0116	0.0071
73	10	85.5	13.73	13.76	13.80	1.00267	0.00266	0.0043	0.0027
73	00	87.5	13.78	13.78	13.78	1.00000	0.00000	0.0000	0.0000

The following notes explain the abbreviated headings in the table.

[a] V_a represents the cubic feet occupied by 1 lb of dry air at the given dry-bulb temperature. The indicated relative humidity does not apply to V_a.

[b] V_m represents the cubic feet occupied by 1 lb of a mixture of dry air and water vapor which has the indicated ET, relative humidity, and dry-bulb temperature. V_m corresponds to the Int. Crit. Tables' definition of specific volume.

[c] V_{as} is Mr. Fleisher's *constant volume*, calculated with less accuracy than in his Table 1.

[d] f_w means the fraction $\frac{\text{total weight of vapor in } V_{as}}{\text{total weight of mixture in } V_{as}}$ or the fraction $\frac{\text{total weight of water vapor in } V_{as}}{W_{as}}$

[e] Aw is the molar fraction of water wapor in the mixture in V_{as}. Taking 18 as the molecular weight of water vapor and 29 as the molecular weight of air, A_w is computed as

$$A_w \text{ equals } \frac{\text{mols of water vapor}}{\text{mols of water vapor plus mols of air}}$$

which equals

$$\frac{(\text{wt. of water vapor})/18}{(\text{wt. water vapor})/18 \text{ plus } (\text{wt. of air})/29}$$

and this equals

$$\frac{f_w W_{as}/18}{f_w W_{as}/18 + 1/29}$$

A_w shows the proportion of water vapor molecules in V_{as}.

All cases are for standard atmospheric pressure, 14.696 lb per sq in.

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[6] Lieutenant Commander (MC), U. S. N., Washington, D. C.

When one considers the manner of heat loss from the body, and the difficulty of ascribing fixed values to the factors of radiation, conduction and convection, and evaporative cooling under controlled conditions, one should not anticipate the existence of any physical expression governing these physiological mechanisms.

The interesting values for specific volume in relation to effective temperature lines are therefore most likely coincidental, and do not inherently prove a fundamental relationship.

The significance of the relationship however is amenable to test. It is possible, for example, to alter barometric pressure from one temperature to one-fifth of an atmosphere, or to substitute a helium-oxygen atmosphere for air. Under these conditions the specific volume concept, if valid, should be applicable as well as at normal barometric pressure.

In conclusion, I should like to state that my attention has been called to the fact that specific volume, as used in the text of the paper, means the cubic feet occupied by a pound of dry air, *and in addition,* its moisture and water vapor. The constant weight of one pound does not include both air and water vapor. This fact complicates the evaluation of the specific volume concept as developed in the paper.

W. H. SANDERS [7] (WRITTEN): Mr. Fleisher has pointed out a purely physical constant associated with the effective temperature or ET line. To understand the relation fully one must realize that this constant, a volume, is occupied by a mixture which varies both in total weight and in ratio of components. So what it takes to make the simple constant is a relatively complex thing after all. This inner complexity is illustrated in Table A, which may be considered a supplement to Table 1 of the paper.

The fourth paragraph, printed inside quotation marks, seems to be incomplete and not wholly in agreement with the text on page 301 of Haas' *Introduction to Theoretical Physics,* Vol. II, second edition, 1929, Van Nostrand.

The last clause of the paragraph containing Equations (1) and (2) appears to have one word wrong. Haas on p. 333 has "——, at given temperature and given pressure, the *densities* of different gases are in the ratios of their molecular weights."

Equations (2) and (3) of the paper conflict, unless R has the same meaning as k. Haas, p. 331, retains the usual significance of R, writing

$$PV = nRT$$

where n represents the number of gram molecules.

The equation

$$\left(\frac{\partial E}{\partial V}\right)_T = 0$$

alone is insufficient for the calculation mentioned in the sentence following it.

The sentence preceding Equation (5) might more accurately read: Including all of the molecules striking face A, the total impulse in time $t = 2x/v$ is

$$Ft = F\frac{2x}{v} = \frac{1}{3} \times \frac{x}{1} N \times 2vm.$$

It might clarify the discussion if the equation immediately preceding (6) were written so as to show the relation between pressure and kinetic energy. Thus,

$$P = \frac{Nmv^2}{3V} = \frac{2}{3}\left(\frac{1}{2}\frac{Nmv^2}{V}\right)$$

[7] Naval Research Laboratory, Anacostia Station, Washington, D. C.

Similarly, Equation (8), where N_1 and N_2 represent number per unit volume, may be written as

$$P = \frac{2}{3}\left(\frac{1}{2}N_1 m_1 v_1^2 + \frac{1}{2}N_2 m_2 v_2^2, \text{ etc.}\right).$$

Putting this in words: The gas pressure equals two-thirds of the kinetic energy per unit volume. Consequently, all perfect gas mixtures at the same pressure have the same kinetic energy per unit volume. (They do not, of course, all have the same kinetic energy per unit mass.) Accordingly, it would seem necessary to interpret the phrases "specific energy of the gas mixture" and "specific kinetic energy" in the two sentences following Equation (9) of the paper as meaning kinetic energy per unit volume.

As a consequence of this simple, fixed relation between pressure and kinetic energy per unit volume, the following gas mixtures will all have the same kinetic energy per unit volume, all being at a pressure of 14.696 lb per sq in.: *a*) dry air at 110 F; *b*) air saturated with water at 73 F; *c*) 21 per cent oxygen, 79 per cent helium at 32 F.

RENÉE V. EULENBERG-WIENER[8] (WRITTEN): I am going to discuss this paper from the viewpoint of the physiologist, moreover from the viewpoint of the molecular minded physiologist and not as a mathematician, engineer or expert in thermodynamics.

The authors have demonstrated, mathematically, that the reactions of man to atmospheric conditions of temperature and humidity expressed in terms of the sensation of comfort or discomfort, arrived at empirically and known as effective temperature, a statistic average, can be related to the constant volume of this mixture of gases, in other words to the number of molecules which bombard the temperature receptor organs in the skin.

First of all what constitutes the feeling of comfort? One may define it as the negative quality of being unconscious of one's body. A resting human being in perfect health is unconscious of its bodily processes. The feeling of discomfort may arise whenever consciousness is directed to any particular function or organ except, of course, during conscious activity. In short, the comfort felt within the effective temperature zone derives from the fact the temperature naturally created within the body as a whole by cellular processes of life, oxidation and reduction, is in equilibrium with that of the surrounding atmosphere. Whenever this balance is displaced, adjustments are called into play and, if they are severe enough or unable to bring about equilibrium, call it to consciousness and create a feeling of discomfort. I said the body as a whole because the temperature within different regions is not alike as it reflects the given intensity of the metabolic processes in each particular tissue or organ.

How is the temperature of the body regulated and what are the means whereby adjustment to environmental temperature is maintained? We know that the skin contains receptor organs for the sensation of heat and cold. The nature of the nerve impulse, that is to say the mechanism of the transport of the stimulus received—the quanta of energy impinging upon the receptor organ, is not agreed upon. Some hold it to be chemical, some in the nature of radiant energy, *i.e.* electrical. But whatever its nature, the impulse, in this case the temperature regulating impulse, travels to temperature regulating centers in the brain where it initiates the proper impulses to be sent to the effector organs which bring about the adjustments necessary for the maintenance of the body temperature within the physiologic range.

These temperature regulating centers have been variously located but the highest center is said to lie within the hypothalimic region of the midbrain. The existence of temperature controlling centers has been experimentally proved and clinically ob-

[8] Ph.D., Tomkins Cove, N. Y.

the ASHVE effective temperature and the authors' effective temperature. The open circles represent points taken from the ASHVE chart for a series of conditions with the same 88 deg ET, and they show a perfect fit with our limit curve for the clothed subject. This coincidence represents an excellent and independent validation of the trend of the ASHVE effective temperature curves at this region of the chart for the clothed subject (and should have been pointed out at a meeting of the Society before this). The closed circles represent a series of conditions with constant kinetic energy or constant volume, *i.e.*, the authors' effective temperature, and quite fortuitously or otherwise their locus falls squarely on our curve for the limits for the nude subject instead of the clothed! I do not, at the present time, feel qualified to give a rational explanation of these coincidences but wish to point out that the agreement between the authors' scale and the effective temperature scale at 73 F, the coincidence of the Pierce curve and the ASHVE effective temperature curve for the upper limits of evaporative regulation for the clothed subject and the coincidence of the authors' scale with the Pierce curve for the limits for the nude subject all point to a common solution.

It should be pointed out that the evidence presented here from our own physiological studies refers only to a warm environment in the zone of evaporative regulation. The independent validation of either the authors' formula or the effective temperature chart under cold conditions will be less simple. Here, again, the relationship for nude and clothed subjects will be distinctly different.

JOHN A. GOFF (WRITTEN): The language and terminology employed by the authors are so confused and confusing that a thermodynamic discussion of their paper is practically impossible. It does appear, however, that two deductions from statistical mechanics may be used as a basis for judging the reasonableness of their proposed correlation. The first deduction is: *in the absence of appreciable intermolecular forces, the pressure of a gas, or of a mixture of gases, is two-thirds of the total translational kinetic energy per unit volume.* The second deduction is: *in a gas, or in a mixture of gases, the total translational kinetic energy per unit weight in mols is proportional to absolute temperature.*

Consider a mixture of dry air and water vapor, namely, moist air at 73 F, 100 per cent saturation and, for comparison, at 88 F, 0 per cent saturation. The following quantities are readily computed for standard atmospheric pressure (29.92 in Hg) using the deductions given.

	73 F, 100 Per Cent	88 F, 0 Per Cent
(1) Weight of water per pound of dry air (lb)	0.0176	0.000
(2) Volume of mixture per pound of dry air (ft^3)	13.80	13.80
(3) Volume of mixture per pound of water vapor (ft^3)	784.3	Infinite
(4) Translational kinetic energy per pound dry air (Btu)	56.31	56.31
(5) Translational kinetic energy per pound water-vapor (Btu)	3200	Infinite
(6) Translational kinetic energy per pound of mixture (Btu)	55.33	56.31

The authors have apparently convinced themselves that the quantity of translational kinetic energy *per pound of dry air* in the surrounding atmosphere (item 4) must determine human comfort. Thus they would say that a human being should feel equally comfortable at 93 F, 100 per cent and at 88 F, 0 per cent. But, according to the first deduction previously stated, the volume of moist air per pound of dry air in it is constant whenever, for a constant pressure, the quantity of translational kinetic energy per pound of dry air is constant. The authors therefore propose volume per pound of dry air as an alternative criterion for human comfort.

The first criticism of the authors' reasoning is that they have omitted to explain why they picked item 4 rather than item 5 or item 6 as a basis for correlation. Furthermore, they do not explain why they completely ignored rotational kinetic energy which is fully two-thirds as much as the translational. But a more fundamental objection is the attempt to use a *property* of a surrounding atmosphere as a criterion for the feeling of comfort of an individual in that atmosphere.

For it appears more reasonable to suppose that the feeling of comfort is brought about by such mechanisms as heat transfer and evaporation in which the speed of these mechanisms is of first importance.

F. C. HOUGHTEN (WRITTEN): The discovery reported in this paper is of great interest to anyone interested in the relation of man to his atmospheric environment, either from the point of view of air conditioning or physiology. The authors have found that two natural laws, namely, the constant volume, air-moisture relationship and the feeling of warmth and many physiological reactions versus air-moisture content relationship agree. The fact that constant volume and molecular bombardment are closely related with each other makes the relationship all the more important for it offers the suggestion that our feeling of warmth and, therefore, some of the most profound physiological processes of the body are functions of the molecular bombardment of its atmosphere.

Accidental coincidences while occasionally met in nature are not common and this discovery is, therefore, one which we cannot ignore or dismiss lightly. If study proves the discovery to be a happy coincidence it will still be an interesting discovery, but of no great importance. However, if on the other hand, it is proved to be a true relationship, the discovery must become of profound importance for it will account for a very important process of life which will offer new avenues of approach to the understanding of our physiological process and to life itself.

ELLSWORTH HUNTINGTON (WRITTEN): I am much interested in the authors' discovery that the constant volume of the air is a good index of its physiological effect. As I understand it, people begin to feel uncomfortable when the constant volume reaches a certain level regardless of how that level is attained. The level may be due to any one of an indefinite number of combinations of temperature and humidity ranging from 77.5 F with saturated air to 98.6 F with no humidity. I do not know how to explain this, but I wonder whether it may not be a step toward the solution of one of the greatest and most neglected of physiological questions. Why do all warm-blooded animals have so nearly the same internal temperature, not far from 100 F?

As yet we seem to have no inkling as to why a temperature of 98.6 F is normal for the body. Practically all mammals have approximately this same body temperature, and that of birds is only a few degrees higher. There is a most extraordinary jump between the temperature of all the cold-blooded forms of life and that of all the warm-blooded types. Between the two types there are only the slightest hints of creatures which preserve an even body temperature at any level much below that of the human body. It would seem as if there must be some physical or chemical reason why the intervening temperatures are undesirable. Or perhaps one might better say, it looks as if some peculiar physical or chemical condition must prevail at a temperature of about 100 F in some such way as at 32 F, although due to far more complex causes. The studies of the Fleishers do not give any hint as to what this reason may be, but they at least take us a step nearer to the solution. I am convinced that if we can discover why both mammals and birds have so high a temperature, and why forms of life with constant body temperatures at any lower level are almost unknown, we shall advance a long way toward solving the most fundamental of all problems of the heating and ventilating engineer.

Another purely incidental point in this paper prompts me to say a word. The authors speak of *irritability in response to the weather or at times of storms.* This

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phrase represents a common viewpoint, but it seems to me far from representing the truth. Or rather, it conveys a misapprehension because it makes no distinction between the different phases of a storm. In my study of deaths day by day in New York City during an eight-year period, one of the most impressive results was the variation in the deathrate at different times during the course of a single storm. When the outdoor temperature rises before a storm, the deathrate is systematically high, regardless of whether the outside temperature is absolutely high or low. At most seasons of the year such a rise in outdoor temperature is the normal prelude to a storm. As a rule it occurs on the day before a storm. It is the time when irritability is felt. While the storm is in progress, that is, after the rain begins, the feeling of irritability generally diminishes and soon vanishes. In fact, many people find that a genuinely rainy day is one of the times when they can work most serenely and steadily. This is especially true when the rain begins to fall the preceding night and ceases during the latter part of the day or in the evening. On such days the mean outdoor temperature normally falls. My studies in New York show that when the temperature thus falls from one day to the next the deathrate also falls.

The third stage in a storm occurs on the day when the actual clearing takes place, or on the day following a night on which such a clearing has occurred. This day, too, even more than its predecessor, is normally characterized by an outdoor temperature lower than that of the preceding day. It is also characterized by a drop in the deathrate. On the clear day after a storm irritability falls to a low level. In other words, people feel good natured. Thus the normal storm is characterized by three stages: 1. a preparatory day with (*a*) rising temperature, (*b*) rising humidity (*c*) falling barometer, (*d*) increased irritability, and (*e*) a high deathrate; 2. a rainy day with (*a*) stable temperature at first with little change from sunrise until a fall begins, (*b*) high humidity at first and then falling humidity, (*c*) more or less stationary low barometer with beginning of a rise, (*d*) lack of irritability with consequent power to concentrate on work, and (*e*) a drop in the deathrate; 3. a day when the air becomes (*a*) cool, (*b*) dry and clear, and when (*c*) the barometer is rising, (*d*) people's spirits are at their best, and (*e*) the deathrate is low.

It seems to me that heating and ventilating engineers might well devote a great deal of attention to discovering what it is which causes the contrast between the oppressive air and high deathrate at the onset of a storm and the stimulating air and low deathrate at its end. The conditions of irritability and health do not seem to correlate well with either temperature or humidity. On the other hand, as I suggested in an address at the Bicentennial Conference of the University of Pennsylvania in September 1940, systematic although very slight variations in the amount of atmospheric ozone seem to harmonize with variations in health and irritability in most of the important cases where these depart from what would be expected on the basis of temperature, humidity, and air motion. At the time when people have a feeling of oppression and irritability just before a storm the atmospheric ozone normally falls to a minimum. Then in a *normal* storm it increases and is at a maximum near the end of the storm at just the time when people have the greatest feeling of exhilaration. Moreover, an annual maximum of ozone occurs regularly in the late winter and early spring. Conditions of temperature cause this season to be almost the most unhealthful time of the year. Nevertheless, the statistics of more than 40 libraries scattered over the United States, Canada, Hawaii, Cuba, Panama, Argentine and Australia all agree in indicating that intellectual activity rises to the highest peak of the year at this season. In other words, in the spring of each hemisphere a surge of unusual intellectual activity coincides with a surge of atmospheric ozone, just as a similar surge of good feeling coincides with a surge of ozone after a storm has become well established and is beginning to pass away.

I am well aware that there is a widespread prejudice against the idea that atmospheric ozone has any special significance. Nevertheless, many otherwise inex-

plicable facts seem to find their most satisfactory explanation only in the hypothesis that atmospheric ozone in extremely minute quantities—less than has been used in practically all experiments—and probably in intermittent doses has a remarkably tonic effect.

In discussing this ozone hypothesis we seem at first sight to have wandered far from the Fleishers' discovery as to the significance of a constant volume of air plus water vapor in producing constant physiological effects regardless of what combination of air and water vapor is involved. This is not the case, however, if we keep in mind that the main aim of the heating and ventilating engineer is to produce a feeling of comfort and to promote health. The importance of the authors' discovery lies in the fact that a comfort line such as 73 deg ET indicates a uniform transfer of heat back and forth between the body and the air. This in turn means a constant degree of stimulus to chemical activity within the body. If there is any truth in the ozone hypothesis, the chemical composition of the air also has an effect upon the degree of chemical activity. Ozone stimulates such activity directly, but it is probably far more important as a catalyser which starts reactions which might not otherwise occur. Thus, if we are to discover the precise conditions which are most desirable for health and comfort, we must take into account not only the emission of thermal radiation from constant volumes of an atmosphere composed of air and water vapor, but the degree to which the chemical activity thus induced within the body is altered by the presence of varying quantities of an active chemical stimulant.[9]

A. P. KRATZ and A. E. HERSHEY (WRITTEN): The authors are to be congratulated on a new and ingenious approach to the subject of comfort. If it is granted that the first premise is valid, then most of the conclusions drawn follow as a matter of course. This premise seems to be that *all points on an effective temperature line from 100 per cent relative humidity to zero per cent relative humidity are on a constant volume line.* The implication of this is that all of the lines of constant volume that can be drawn on the chart shown in Fig. 1 not only actually do, but also necessarily must, coincide with corresponding lines of constant effective temperature. These lines of constant volume evidently are practically parallel to one another, and the ones in the immediate neighborhood of 75 deg ET practically coincide with the constant effective temperature lines. Above 75 deg ET the slopes of the constant effective temperature lines are greater than those of the constant volume lines, while for below 75 deg ET they are less than those of the constant volume lines. That is, if lines of constant volume are drawn through the intersections of the effective temperature lines and the saturation curve, the deviations between the lines of constant effective temperature and the lines of constant volume become successively greater progressing in either direction from the 75 deg ET line. For example, the constant volume line passing through 86.5 deg ET at the saturation curve intersects the base line of the chart at 83 deg ET instead of at 86.5 deg ET. Similarly a constant volume line drawn through 50 deg ET at the saturation curve would intersect the base line of the chart at 55 deg ET instead of at 50 deg ET.

It would seem that these consistent and progressively increasing deviations can hardly be explained away as errors in the effective temperature lines. It is barely possible that the agreement in the optimum comfort range may be due to the fact that the slope of all of the lines of constant volume happens to be the same as that of the 75 deg ET line, rather than that there is a functional relation between the constant volume and the feeling of hot and cold. Furthermore, following the reasoning employed in the paper, it would seem that the effective temperature for optimum comfort for both summer and winter should be 75 F, since the best agreement between the lines of constant effective temperature and constant volume occurs at this effective temperature.

[9] The ozone hypothesis is discussed more fully in a paper entitled, Climatic Pulsations and an Ozone Hypothesis of Libraries and History, by Ellsworth Huntington, in *Conservation of Renewable Natural Resources*, University of Pennsylvania Bicentennial Conference, Philadelphia, 1941, p. 99.

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V. L. SHERMAN (WRITTEN): Whether or not it is fortunate that I have been asked to discuss this paper remains to be seen. The opportunity is greatly appreciated.

At any time within the last four years I would have agreed that our sense of comfort, or discomfort, is a function of the kinetic energy of the surrounding atmosphere. I had not, however, associated this with the effective temperatures.

The effective temperature lines have been of the greatest value to engineers in the promotion of wholesome progress, but in my belief no normal person exists. The lines then become the normal of a large group. That these lines come so close to the constant volume lines does not seem strange, now that my attention is called to it.

May I be permitted to disagree with the statement of the authors that "As soon as the constant volume line at zero per cent relative humidity corresponds to the blood temperature, the entire physiological reaction of the body changes." I believe entire changes have taken place before this state is reached, and will continue afterwards. Any peak in the rate of change depends upon the individual. The same criticism holds for the statement at the end of the paragraph in which it is said "a feeling of warmth and body perspiration take place, all indicating the *beginning* of a physiological change." As engineers we are prone to ask for exact points.

If I am willing to agree in general, and it is not difficult, perhaps some of my own reasons should be given. Although I have no proof, my feeling has been that our sense of comfort or discomfort is dependent upon the action within the lungs, a subconscious index. As atmospheric conditions grow more severe we are aware of troubled breathing. As a young man working in ships' engine rooms in hot atmospheres I could obtain relief by merely putting my face to an open port.

The average human body has a surface, we will say, of 20 sq ft. The whole surface of the walls of the alveoli are about 1,000 sq ft. During one minute 3 to 7 litres of blood, an average of 300 cu in., pass through the lungs while the body is at rest. At exercise an average of 25 litres, 1,500 cu in., is properly taken care of with reasonable surrounding atmosphere. About 50 cu centimeters of carbon dioxide per litre is picked up in transit by the atmosphere in exchange for as much or a little more oxygen. About 3 cu in., for every 60.

Going into the lungs the air may be filtered, moistened, and altered in pressure. The kinetic energy of that air is responsible for what takes place within the lungs. If there should be a shift in this action within the lungs some form of compensation will exhibit itself in some other part of the body. This will not be entirely in the region of the skin. But we have used skin conditions as an index so long, and the other parts are so inaccessible. The skin's function may be changed from insulation to radiation, and to radiation and evaporation.

Getting away from the normal human, consider the disabled individual, the heart-asthma-thyroid creatures that pray for a reasonable atmosphere. One doctor friend of mine was relieved of his asthmatic condition simply by sleeping in an air conditioned room. (It might be mentioned in passing that in doing this he was subject to headache during the following day unless he allowed the room air to approach outside air conditions some time before arising. I shall not attempt to give the medical explanation.)

Where air conditioned rooms were available many critical cases have been pulled through, even, I believe, heat exhaustion. All of these cases, I am sure, would have experienced nearly as much relief if they had merely breathed the conditioned air.

This paper has pointed out to me a number of relations of which I had thought but little. I hope the authors will accept my highest compliments and thanks for their work.

C. TASKER (WRITTEN): This paper presents an explanation of the effective temperature scale as ingenious as most of us would have expected from the senior

author, who will doubtless defend it with his accustomed vigor against all comers. The paper contains much food for thought and my only purpose in commenting is to provoke discussion.

The Society's present effective temperature scale takes no account of radiation or rather, I should say, assumes that the mean radiant temperature of the surrounding surfaces is equal to the dry-bulb temperature of the air, and yet, as is shown by other papers presented at this meeting, radiation plays an important part in comfort. Where does radiation enter into the author's explanation of the reasons for comfort at widely differing air conditions?

Taking the figure of 73 deg ET the dry-bulb temperature can vary from 76.1 deg to 81.7 deg over the range of humidities shown in Table 1. Now, if the mean radiant temperature is around 81 to 82 deg and the air is at the same dry-bulb temperature with a relative humidity of 30 per cent, the temperature will be about 73 deg ET and most people would, under normal summer conditions, be comfortable. However, if the mean radiant temperature dropped slightly to around 80 deg and the air temperature were changed to 76 F, and for purposes of argument let us assume it could happen fairly quickly, theoretically, so long as the humidity was increased to around 70 per cent, most persons would indicate no change in their feeling of comfort. However, according to a previous paper,[10] if there was a difference of 4 degs between the mean radiant temperature and the air temperature, the effect would be equivalent to a maximum of 2 deg ET and people would feel warm, since the condition would be more nearly equivalent to 75 deg ET than 73 deg ET. There would be no actual change in constant volume and yet people would feel warmer as if there were a change, and would continue to do so until the mean radiant temperature had been reduced to the air temperature, when they would again feel comfortable, still with no change in constant volume.

Until tests can be conducted to verify this theory and the suggested changes in the slope of the effective temperature lines, which, as I understand it, were never supposed to be very accurate below 20 per cent relative humidity and above 85 per cent relative humidity, effective temperatures will probably remain purely empirical but very useful guides to comfort.

AUTHORS' CLOSURE: This paper has received so much general and theoretical discussion, in many cases not carefully studied or in other cases misinterpreted, that we are desirous of simplifying and again justifying this paper.

The theory is dependent upon the fundamental thermodynamic equation applying to perfect gases

$$pV = RT \quad \ldots\ldots (1)$$

Where p = absolute pressure in pounds per square inch corresponding in ordinary pressure to the barometric pressure, or is the pressure exerted by a unit volume of any gas

V = unit volume

R = gas constant

T = absolute temperature

and a second equation

$$p = \tfrac{1}{3}nmv^2 \quad \ldots\ldots (2)$$

Where p = pressure in pounds per square inch exerted by any gas

n = Avogadro's constant, that is, the number of molecules in a unit volume

m = mass of a single molecule of the gas

v^2 = average of the squares of the molecular velocity

[10] ASHVE RESEARCH REPORT No. 1174—Comfort Requirements for Low Humidity Air Conditioning, by F. C. Houghten, H. T. Olson and S. B. Gunst. (ASHVE TRANSACTIONS, Vol. 47, 1941.)

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From these equations and laws and from the general thermodynamic laws of gases, that is, Boyle, Charles, Guy-Lissac and Dalton
Substituting

$$P = P_1 + P_2 + P_3 \text{ etc.} \quad \ldots\ldots (3)$$

Equation (2) in Equation (3)] $P = (\frac{1}{3}n_1m_1v_1^2) + (\frac{1}{3}n_2m_2v_2^2)$ etc. $\ldots\ldots (4)$

$$\text{If } V_t = V_a + V_w \quad \ldots\ldots (5)$$

Where V_a = volume of the air molecules in the total unit volume

V_w = volume of the water molecules in the total unit volume

Multiplying Equation (4) by Equation (5), you can get

$$P_tV_t = (V_a + V_w) \times (\tfrac{1}{3}n_1m_1v_1^2) + (\tfrac{1}{3}n_2m_2v_2^2) \quad \ldots\ldots (6)$$

Simplifying

$$P_tV_t = \tfrac{1}{3}N_am_av_a^2 + \tfrac{1}{3}N_wm_wv_w^2 \quad \ldots\ldots (7)$$

Where N_a = total molecules of air involved

N_w = total molecules of water involved

$\therefore$ If P_t = a constant C and V_t = a constant C $P_tV_t = C$

$\therefore$ Kinetic Energy of the Equation = a constant

The authors also, according to Maxwell and other physicists, can show that for two different gases at the same temperature the average kinetic energy per molecule of the first gas must equal the average kinetic energy per molecule of the second; and that, referring again to Avogadro's law, at the same constant temperature and pressure the number of molecules in a unit of space is the same, irrespective of the character of the molecule. Also, from these equations, it can be seen that the kinetic energy is a function of the absolute temperature.

We claim that in equal volumes, with varying temperatures, the kinetic energy created by the action of the mixture is equal, irrespective of the relationship of one kind of molecule to the other—in other words, as long as the volume of the combination of air and water vapor remains constant, the kinetic energy is constant. On our lines, the assumption is, that you have two points of constant volume: one, a pound of dry air, the other, the same pound of dry air, with the water vapor to saturate it; that the kinetic energy in the dry air is a certain amount, and that the kinetic energy in the lower temperature saturated air is the same. With a change in temperature from one point to the other, the total volume remaining the same, the energy of the air molecules, decreasing from the 0 per cent saturation point with a drop in temperature, are compensated for by the energy of the water vapor molecule, making up the increased volume essential to maintaining a constant volume. In our original paper we talk of specific volume, but in psychrometry the specific volume is understood to be cubic feet per pound of the mixture. Our theory is based on *constant volume* in cubic feet per pound of dry air, plus the moisture, in other words, a constant volume line based on the *mixture*.

Along a line starting at 73 deg saturation, and the saturation point is the basis of the effective temperature line, and continuing on a line of constant volume with decreasing water vapor and increasing temperature, the effective temperature line developed by the Society corresponds almost exactly with our constant volume line. We feel this is because between 71 and 73 deg more people were comfortable than under any other condition, and, consequently, the interpretation of the reactions on which the effective temperature lines were based was more nearly accurate. The deviation from the constant volume line was much less than the difference of opinion of the subjects on whose expression of opinion the effective temperature line was based.

There hardly seems to be a possibility of analysing the relationship from any other angle than the bombardment of the molecules in the particular area surrounding the subject.

As we move out of the comfort zone, we find a wider discrepancy between the authors' lines and the effective temperature lines; but a careful examination of the detailed results of the Laboratory in developing their lines would indicate that it would have been just as logical to assume different points at which to develop their lines as it would have been to accept the lines which they have indicated. A variation of ½ or ¾ deg in the body temperature of the subjects at the higher effective temperatures made a great difference in the location of the points on the curves, and there was no stability of results within this narrow range. All field tests indicated to a marked degree the close relationship of the effective temperature reactions and the constant volume lines indicated as the Fleisher energy lines.

There is another definite reason for the effective temperature lines at uncomfortable conditions varying from the constant volume lines of the Fleisher theory. These lines deviate most at higher dry-bulb temperatures, and the higher dry-bulb temperatures must necessarily have resulted in higher enclosure temperatures of walls, ceilings, floors and equipment, which created a radiation effect which was not accounted for or taken into consideration in the Laboratory tests. The high temperature conditions were only investigated in the Laboratory and not in the field.

The other point we want to bring out is, that all field tests and all Laboratory tests, covering the investigation of thousands of people, indicated that at 77½ deg ET no one expressed himself as comfortable, unless his temperature, under normal conditions, was above the so-called normal or standard temperature of practically all human beings, 98.6 deg; and the 79 deg Fleisher line crosses the 0 per cent relative humidity line at almost exactly the blood temperature. This would seem to indicate that the whole theory was not coincidental; that human beings definitely react to the kinetic energy bombardment of the molecules in the cubicles of air surrounding them, and that the body temperature is a result of the survival of the fittest; that over thousands of years, or tens of thousands of years, those people who could not withstand, for a definite length of time, repeated day after day for probably hours or days, the bombardment corresponding to the kinetic energy encompassed in the relationship of air and water molecules from 79 deg saturated, to 98 deg with 0 per cent relative humidity, had died off, and the human body has finally adjusted itself over these thousands of years to this bombardment as a maximum condition for normal equilibrium.

The idea that has been expressed by certain people, that human beings react or maintain their equilibrium on a constant total heat line, are obviously mistaken, because everything would indicate (not only from our theory, but from all of the experimental work that was done) that a constant heat line, as developed on our psychrometric chart, has no relationship to equal comfort or equal reaction; that starting at the higher saturation temperatures, a constant enthalpy or total heat line swings into the higher temperature so rapidly, that our research indicates very definitely that there is no equal reaction or equal comfort on these lines.

There is a second phase to this theory, which was not brought out in the paper, although it was mentioned. We are dealing with the subject from one side of the equation. We have shown that, seemingly, our equilibrium is maintained constant as long as the sum of the molecular bombardment of different types of molecules remains the same, and this bombardment can readily be calculated in terms of energy, since we know from research over a long period of time the mass of the molecule and its velocity. We can also determine the effect of radiant energy when added to the energy of the air and water vapor.

The check on the problem is to determine whether the energy given out by human beings at the temperature of the body, most of which is radiated, is equal to the

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energy coming to the body from outside. This should not be too difficult a problem. If we assume that the body is a black body; if we measure the metabolism of the body under different conditions and from these determine the energy equation for either the mass or surface of the body, we should, if we go about it accurately enough, definitely settle whatever controversy might exist as to the reliability of the theory which the authors have set up. The John B. Pierce Laboratory in New Haven has made investigations of the reactions of human beings to their environment under conditions that would check our theory, and have expressed themselves as being in definite sympathy with the theory and the results we have indicated. The two points of discrepancy, which probably are not points of discrepancy at all, but points of missing data, have to do with the change in the so-called effective temperature lines with varying air motion. This can probably be explained by the theory that evaporation from the body adds water vapor molecules to the air close to the surface of the body, and this would increase the kinetic energy of the air surrounding the body and consequently impose a greater bombardment on the body, itself. Under high air movement, two things would take place. The moisture-laden air, having a higher kinetic energy, would be carried away from the body, and, secondly, increased evaporation would have a tendency to lower the dry-bulb temperature, thus decreasing the so-called effective temperature surrounding the body, and produce in that way easier equilibrium. There is no doubt that air movement more rapidly changes the surrounding conditions of the air and, consequently, works a change in the quantitative results, without in any way affecting the basic equation.

In connection with the written discussion by Mr. Sanders, the authors are very grateful for his careful analysis of the equations and for the tables which he has furnished. They apologize for having, to a certain extent, abbreviated Haas' text, which they took from p. 301 of the *Introduction of Theoretical Physics,* by Van Nostrand; also, for having abbreviated the development of the formula to arrive at the same final equation as that arrived at by Mr. Sanders. The authors do not want to comment on conclusions until they have discussed Dean Goff's comments in his discussion, which they have very carefully noted.

In the authors' closure they have simplified the development of the kinetic energy equation to reconcile it to the basic kinetic equation for energy, represented by the movement of the molecules in a gas at a definite pressure. In their original paper, they attempted to reconcile the equation as developed by Haas in his *Introduction to Theoretical Physics,* with the development of the equation to the commonly accepted kinetic energy equation of gases in which the number of molecules, the mass of the molecules and the mean of the squares of the velocity were involved. The authors believe that the simplified equation which they have inserted removes the objections that have been stated.

Dean Goff has stated two deductions as a basis for judging the reasonableness of the authors' correlation. The authors see nothing in the two deductions which in any way would conflict with the equations that they have arrived at. The quantities computed by Dean Goff for standard atmospheric pressure of 29.92 in. of water, using the foregoing deductions, indicate that at 73 F, 100 per cent relative humidity, and 88 F, 0 per cent relative humidity, the *volume* of the mixture per pound of dry air is equal at 100 per cent saturation and 0 per cent saturation at the two different temperatures—in other words, 13.80. He has also shown that translational kinetic energy per pound of dry air with 100 per cent saturation at 73 F and at 0 per cent relative humidity, at 88 F is 56.31 in both cases. Dean Goff will probably find that at any point along a 13.80 volume line, between the two limiting points of 100 per cent saturation and 0 per cent saturation, the translational kinetic energy per pound of air would remain 56.31. It was because of this exact relationship and because of the fact that at many points along this particular line the sensations expressed by people in contact with this condition were the same, that the authors assumed a relationship which certainly was not predicated on the fact that the

volume along this particular line was constant. It was the fact that a constant reaction along this line had been developed not by them, but by the Society, over many years, that interested them to the extent of evaluating the reason. Having a constant volume as a basis, they felt very definitely that the relationship of the volumes and pressures could indicate only one thing, and that was a constant kinetic energy at all points along this line. The authors do not contend, as Dean Goff imagines, volume per pound of dry air as an alternate criterion for human comfort. They claim that the energy that is involved in the constant volume is the energy coming from outside to create the condition to which the human being reacts.

Further commenting on Dean Goff's discussion, the authors have purposely omitted to discuss the rotational kinetic energy which is inherent in the volume of air which is under discussion. The assumption would necessarily be, if, as Dean Goff says, the rotational kinetic energy is two-thirds as great as the translational kinetic energy, that it then should also be equal at all points along this line, as the translational kinetic energy is equal along this effective temperature or equal volume line. Consequently, if the rotational kinetic energy has any effect on interposing surfaces and is not entirely dissipated in intermolecular reactions, then it would follow that it was equal at all points where the translational kinetic energy was equal, according to Dean Goff's own statements. It might be, and the authors make this statement as a suggestion, that with a rise in temperature, the rotational energy or, according to Equation (2) in Chapter I of the 1942 Guide, where

$$pv = BT - A(T)p$$

would decrease the kinetic energy, that is the kinetic energy that would affect the surfaces against which the bombardment was carried out. This might possibly be interpreted as a reason for the empirical effective temperature lines at the higher temperatures tending to a higher point than the kinetic energy lines on constant volume would indicate in the tests of the Society.

But where the authors take the greatest exception to Dean Goff's comments and where they think that he has completely misunderstood the philosophy of the theory is in his "fundamental objection . . . attempt to use a *property* of a surrounding atmosphere as a criterion for the feeling of comfort of an individual in that atmosphere." All of the work that we of the Society in our research have attempted has been based on the reactions of human beings to their environment. They are in balance or equilibrium with that environment, and their own mechanism of heat transfer and evaporation is primarily an adjustment to the particular sea in which they live. The losses by heat transfer and evaporation are only the variables that compensate for the kinetic energy coming to them from outside themselves and making it essential for them to rid themselves of the products of combustion which maintains a normal or so-called normal temperature within their bodies. When conditions in the environment exceed the kinetic energy level which the body is radiating, a rise in the temperature of the body occurs and remains at the constant higher point as long as the increased bombardment from without persists.

The authors would like to refer to one additional remark of Dean Goff, in which he asked for the reason of the authors for not orienting their conclusions and their equations against his item 6, that is, the translational kinetic energy per pound of the mixture. As he, himself, shows, the mixture not only varies in its translational kinetic energy on this line, but he will find that it also varies greatly in its volume. This, in the opinion of the authors, is as it should be. Given a constant barometric pressure, a human body is bathed in a certain quantity or certain volume of air and not in a certain weight of air, and the kinetic energy in the constant volume is constant, and the kinetic energy in the varying volume, based on pounds of the mixture, is not constant.

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The authors see nothing in Dean Goff's discussion which, in their opinion, detracts in any way from the fundamental basis of their theory.

In connection with Mr. Huntington's discussion, the authors are glad to observe the recognition which he gives to their discovery. The authors felt, and still feel that their discovery simply opens the field to much wider investigation and that any energy quanta coming from the surrounding medium could be additive in developing divergences from a predicted reaction.

No. 1177

SUMMER COOLING IN THE RESEARCH RESIDENCE WITH A GAS-FIRED DEHYDRATION COOLING UNIT

By A. P. KRATZ,* S. KONZO** AND E. L. BRODERICK,*** URBANA, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Gas Association* conducted in the Research Residence of the *National Warm Air Heating and Air Conditioning Association* and the University of Illinois.

DURING the summers of 1932 to 1938 inclusive, extensive studies on summer cooling in the Research Residence have been conducted with the use of ice as a medium for cooling water circulated through cooling coils, with water at three different temperatures used directly through cooling coils, and with mechanical condensing units expanding the refrigerant directly into cooling coils. In the course of the investigations two different sizes of condensing units were used, and studies were made both with and without introducing cool outdoor air into the house at night. These studies on summer cooling were extended in 1940 to include the performance and operating characteristics of a gas-fired, dehydration, cooling unit.

DESCRIPTION OF COOLING EQUIPMENT

The Research Residence together wtih the forced-air heating system has been described in previous publications.[1, 2] During the summer of 1939 the exposed sidewalls and ceilings of the Residence were fully insulated with mineral wool. The calculated cooling load for the structure based on design conditions of 95 F outdoor dry-bulb temperature, 76 F outdoor wet-bulb

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[1] ASHVE Research Report No. 1034—Study of Summer Cooling in the Research Residence Using Water from the City Water Mains, by A. P. Kratz, M. K. Fahnestock, S. Konzo and E. L. Broderick. (ASHVE TRANSACTIONS, Vol. 42, 1936, p. 193.)

[2] Investigation of Summer Cooling in the Warm-Air Heating Research Residence, by A. P. Kratz, M. K. Fahnestock and S. Konzo. (University of Illinois Engineering Experiment Station Bulletin 290, January 1937.)

Presented at the Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., January 1941.

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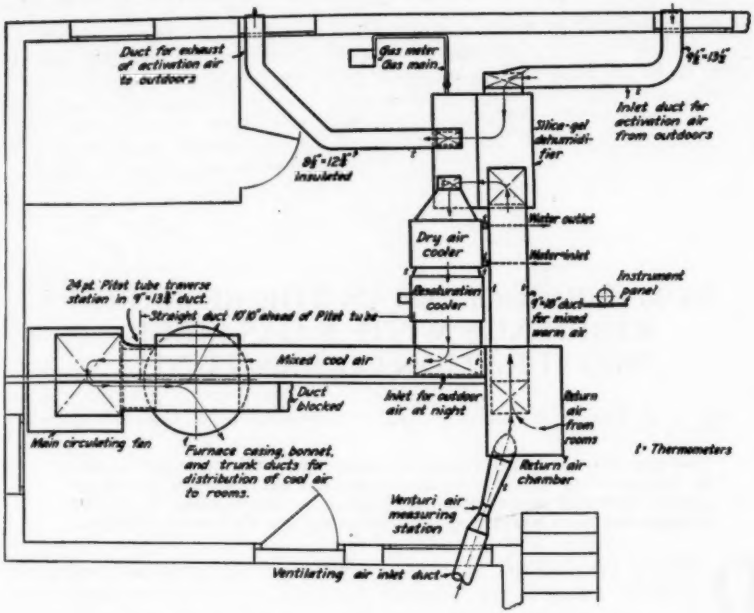
Duct for exhaust of activation air to outdoors
Gas meter
Gas main
Inlet duct for activation air from outdoors
Silica-gel dehumidifier
Water outlet
Dry air cooler
Water inlet
Instrument panel
Resaturation cooler
24 pt. Pitot tube traverse station in 9'-13⅜" duct.
Straight duct 10'0" ahead of Pitot tube
Mixed cool air
Duct blocked
Inlet for outdoor air at night
Return air from rooms
Thermometers
Main circulating fan
Furnace casing, bonnet, and trunk ducts for distribution of cool air to rooms.
Return air chamber
Venturi air measuring station
Ventilating air inlet duct

FIG. 1. PLAN OF COOLING PLANT AND DUCT ARRANGEMENT

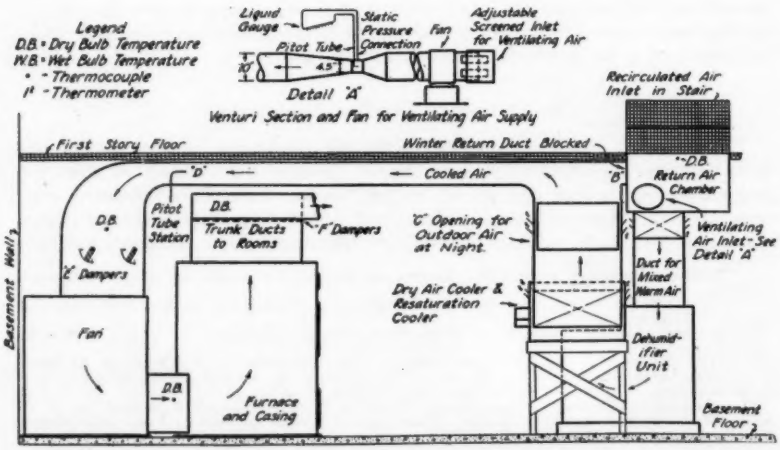

FIG. 2. ELEVATION OF COOLING PLANT AND DUCT ARRANGEMENT

temperature, 80 F indoor dry-bulb temperature, 50 per cent indoor relative humidity, and one air change per hour brought in for ventilation, amounted to 25,300 Btu per hour. Similarly, the calculated cooling load for the uninsulated structure was 37,500 Btu per hour. Hence, by completely insulating the Residence the calculated cooling load was reduced from 37,500 Btu per hour to 25,300 Btu per hour or a reduction of 32.5 per cent.

The total space cooled, consisting of the first and second stories of the Residence, amounted to 14,170 cu ft.

The arrangement of the distributing ducts and fan in the forced-air system was similar to that used in previous investigations.[3] Fig. 1 shows the plan and Fig. 2 shows the elevation of the cooling unit and connected ducts. On the suction side of the circulating fan provision was made for taking in 1500 cfm of air from outdoors at night. The cooling unit was composed of three separate parts, namely the gas-fired silica-gel dehumidifier, the dry air cooler, and the resaturation cooler. As shown in Fig. 3, in the first stage of the cooling and dehumidification process the recirculated air entering the stair grille joined with the ventilating air from outdoors and the resulting mixed warm air entered the dehumidifier. This air in passing through the silica-gel contained in the rotating drum, located in the dehumidifier, was dried and heated by adsorption. The silica-gel in the rotating drum was continuously reactivated, and the moisture was taken up by the activation air brought in from outdoors. The heat required for this process was supplied by a gas burner supplementing the heat contained in the activation air.

In the second stage the heated dry air leaving the dehumidifier passed through the dry air cooler, or water cooled coil, and the dry-bulb temperature was reduced from about 130 F to 78 F.

In the final stage of the cooling process the air leaving the dry-air cooler was subjected to evaporative cooling in the resaturation cooler, resulting in a decrease in dry-bulb temperature and an increase in moisture content. The resaturation cooler contained a rotating wet screen through which the air was passed.

The conditioned air, which was materially cooler and dryer than the air entering the cooling unit, then passed through the furnace casing into the ducts of the air distribution system and was delivered out of the registers into the rooms to be cooled. A typical flow diagram, giving the temperatures and humidities at each stage of the cooling process, is shown in Fig. 3.

It may be observed from the values given in Fig. 3 that the final temperature and humidity of the conditioned air were directly dependent upon whether or not the resaturation cooler operated. When the resaturation cooler was in operation the dry-bulb temperature of the air leaving the cooling unit was about 64 F and the moisture content was about 62 gr (grains) per pound of dry air. However, when the resaturation cooler was not in operation, the dry-bulb temperature of the air leaving the cooling unit was about 78 F and the moisture content was about 47 gr per pound of dry air. Hence, by operating the resaturation cooler in conjunction with both the dehumidifier and the dry air cooler, very cool but somewhat dry air was obtained as an end product, whereas by operating only the dehumidifier and the dry air cooler, slightly cool but very dry air was obtained.

To a limited extent it was possible to control the humidity of the air in

[3] Loc. Cit. Notes 1 and 2.

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a given space independently of the dry-bulb temperature. In this connection it is of interest to note that with either the mechanical condensing unit or with water cooled coils, the humidity maintained in the house was primarily dependent on the temperature of the cooling coil,[4] and it would not have been possible to obtain independent control of the humidity and air temperature without resorting to some form of reheating.

The automatic control system which coordinated the operation of the resaturation cooler with that of the dehumidifier and dry air cooler included

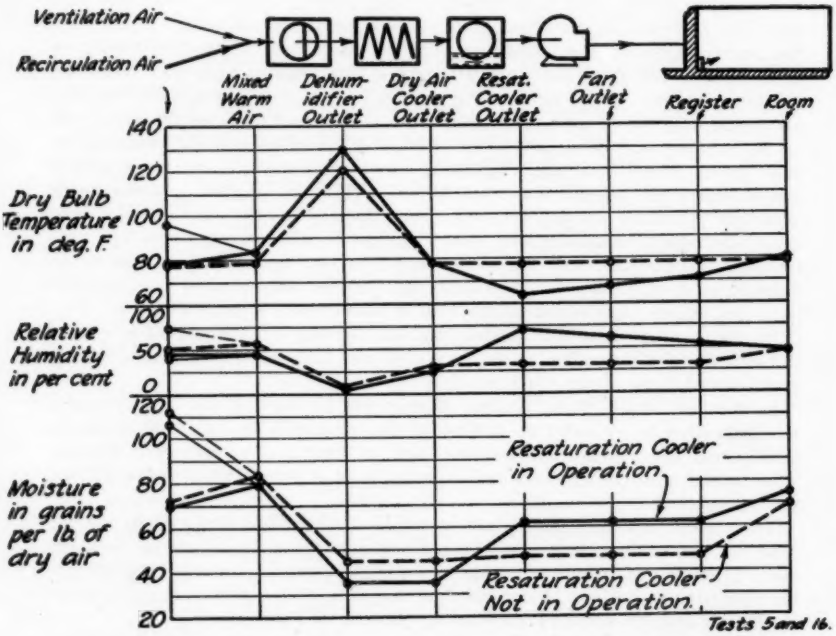

FIG. 3. FLOW DIAGRAM FOR AIR PASSING THROUGH COOLING SYSTEM

a room thermostat and room humidistat, both of which were located in the second story hall. A simplified wiring diagram of the control system is shown in Fig. 4. The four possible combinations of temperatures and humidities of the air that might occur in the Residence and the manner in which the controls would respond in each case are listed in Table 1. For example, as shown in case *A* in Table 1, when the dry-bulb temperature was higher than the setting of the room thermostat and at the same time the relative humidity was higher than the setting of the room humidistat, all three elements of the cooling unit would be brought into operation.

[4] Summer Cooling in the Warm-Air Heating Research Residence with Cold Water, by A. P. Kratz, S. Konzo, M. K. Fahnestock, and E. L. Broderick. (University of Illinois Engineering Experiment Station Bulletin 305, p. 44, August 1938.)

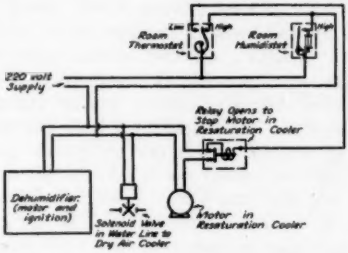

FIG. 4. SIMPLIFIED WIRING DIAGRAM FOR COOLING PLANT

For the purpose of these tests the temperature of the water supplied to the coils in the dry air cooler was maintained at 75 F which was representative of water temperatures commonly available. Since the temperature of the local water supply was about 58 F, considerable auxiliary equipment shown in Fig. 5, which would otherwise have been unnecessary, was required in order to furnish approximately 300 gal of water per hour supplied to the dry air cooler at a temperature of 75 F.

METHOD OF CONDUCTING TESTS

Continuous records were made during the summer of the temperatures of the air in the various parts of the house and of the outdoor air. During the periods of operation, observations were made of the quantity and temperatures of the water passing through the dry air cooler, the gas consumption in the dehumidifier, and the electrical inputs to the motors operating the cooling unit and the circulating fan. Observations were also made of the temperatures and humidities of the air in various sections of the duct system and in the house.

During all of the tests the windows in the attic remained open, and those on the first story remained closed. The recirculated air was returned through the return air grille located in the stair risers in the first story hall. With

TABLE 1—CONTROL OF COOLING UNIT BY THERMOSTAT AND HUMIDISTAT

CASE	ROOM TEMPERATURE RELATIVE TO THERMOSTAT SETTING	ROOM HUMIDITY RELATIVE TO HUMIDISTAT SETTING	PARTS OF COOLING UNIT IN OPERATION
A	High	High	1. Dehumidifier 2. Dry Air Cooler 3. Resaturation Cooler
B	High	Low	1. Dehumidifier 2. Dry Air Cooler 3. Resaturation Cooler
C	Low	High	1. Dehumidifier 2. Dry Air Cooler
D	Low	Low	None

the exception of one series of tests (*Series 4-40*), artificial cooling during the day was supplemented by cooling with outdoor air at night. For this purpose the attic door and 11 windows on the second story were opened in the evening when the effective temperature outdoors became equal to that indoors. At this time the basement door was opened and the dampers were set so that the fan delivered outdoor air, equivalent to 6.4 air changes per hour, through the duct system to both stories. With one exception, air from outdoors was thus supplied until 7:00 a.m., when the windows and doors were closed, the dampers were reset, and the fan was stopped. In *Series 2-40* outdoor air was supplied only up to 6:00 a.m. instead of 7:00 a.m.

When the cooling unit was in operation, the total quantity of air delivered by the circulating fan was 804 cfm, equivalent to 3.4 air changes per hour.

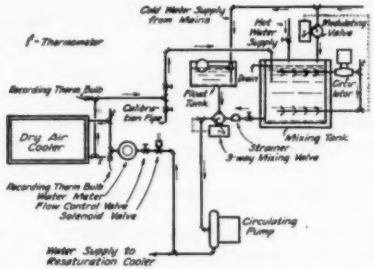

FIG. 5. AUXILIARY EQUIPMENT FOR MAINTAINING TEMPERATURE OF WATER TO DRY AIR COOLER

Of this amount, 232 cfm, equivalent to one air change per hour, represented the outdoor air admitted for ventilation.

The Residence was operated strictly on the schedules described in connection with the following discussion of the different test series:

Constant Effective Temperature. Cooling Supplemented by Outdoor Air at Night. (*Series 2-40.*) In this series of tests, cooling with outdoor air at night was stopped at 6 a.m. During the period between 6 a.m. and the start of the test the recirculating and ventilating fans were not operated. When the effective temperature in the house rose to 74 deg the cooling unit was manually started. The ventilating fan was also started and was operated during the entire period of the test. Before the cooling unit was started, the prevailing relative humidities ranged from 60 to 75 per cent, and at the time of starting, the indoor dry-bulb was between 77 and 78 F. Conditions at the start of the test thus usually corresponded with case *C* shown in Table 1. After the start of the test the cooling unit was operated automatically by the controls. The room thermostat was set to maintain 80 F dry-bulb temperature and the room humidistat to maintain 50 per cent relative humidity, corresponding to an effective temperature of approximately 74 deg. When the cooling plant operated intermittently both the main circulating fan and the cooling unit were started and stopped at the same time.

Constant Dry-Bulb Temperature. Cooling Supplemented by Outdoor Air at Night. (*Series 3-40.*) Cooling with outdoor air at night was stopped at 7:00 a.m. This series differed from *Series 2-40* in that the tests were not started until the dry-bulb temperature rose to 81 F instead of to a dry-bulb temperature corresponding with an effective temperature of 74 deg. The cooling unit was started manually and thereafter the room thermostat and humidistat maintained a dry-bulb temperature of approximately 80 F and a relative humidity of approximately 50 per cent. With the exception of a few minor details of operation, these tests were exactly comparable with those in *Series 3-34* previously conducted [5] in the uninsulated house, and for which a mechanical condensing unit was used.

Constant Effective Temperature. Cooling Not Supplemented by Outdoor Air at Night. (*Series 4-40.*) For this series of tests no cooling was done with outdoor air at night. The house remained closed day and night and the plant was started when the indoor effective temperature rose to 74 deg, after which it operated automatically with the room thermostat set at 80 F and the humidistat set at 50 per cent relative humidity. Outdoor air for ventilating purposes amounting to one air change per hour was introduced into the house both day and night.

Low Relative Humidity. (*Series 5-40.*) During a 48 hour period of hot weather the plant was operated with the controls set to maintain approximately 83 F and 30 per cent. The plant was started when the indoor effective temperature rose to 74 deg and was operated automatically by the controls during the test period.

RESULTS OF TESTS

Constant Dry-Bulb Temperature. Cooling Supplemented by Outdoor Air at Night. (*Series 3-40.*) The tests comprising *Series 3-40* were made in the insulated structure under operating conditions duplicating as closely as possible those maintained for the tests made in 1934, at which time a mechanical condensing unit was used to cool the uninsulated structure. As far as comfort is concerned, this method of operation was subject to the same limitations as those observed in connection with the tests made with mechanical refrigeration and with water, which have been discussed in previous publications.[6] General data and results from all of the tests made in 1940 are shown in Table 2.

The operating characteristics of the cooling plant can best be illustrated by the results obtained on a typical day. For this purpose the test made on August 14, 1940, was selected as typical of the operation obtained on a warm day, and the results are shown in the first column of Table 3 and in Fig. 6. The data presented in the table were obtained at 6:10 p.m., at which time only the dehumidifier and dry air cooler were operating. On mild and medium warm days long periods of time frequently occurred during which very little cooling was necessary, and the action of the dehumidifier and dry air cooler was sufficient to maintain the required dry-bulb temperature in the house without being supplemented by that of the resaturation cooler.

The air conditions at different locations are given in items 5 to 17. The temperature of the air leaving the registers was approximately 78.3 F and the

[5] Loc. Cit. Note 2.
[6] University of Illinois Engineering Experiment Station Bulletins Nos. 290 and 305, pp. 100 and 41 respectively.

TABLE 2—COMPLETE DATA AND RESULTS FOR TESTS—1940

DATE 1940	TEST No.	SERIES No.	WEATHER DATA								INDOOR AIR CONDITIONS				TEMPERATURE DIFFERENCE		OPERATION OF THE FAN			
			Degree Hours			Total Sunshine in Hours and Min.	Outdoor Air				At End of Night Cooling		Avg. During Period Requiring Cooling				Night Air Cooling Previous to Test Period		Recirculation During Test Period	
			Above 80 F	Above 85 F	Above 90 F		Maximum Temp. During Test, F	Minimum Temp. Previous to Test, F	Avg. During Period Requiring Cooling						Outdoor Max. to Avg. Indoor F	Avg. Outdoor to Avg. Indoor F	Duration Hours	Power Kwhr.	Duration Hours	Power Kwhr.
									Temp. F	Hum., Per Cent	Temp. F	Hum., Per Cent								
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
7–21	1	2–40	75.8	29.5	4.3	13:09	92	66	86.2	56.3	76.6	66.4	80.4	48.8	11.6	5.8	12.00	7.0	6.68	5.8
7–22	2	2–40	80.0	31.2	4.3	14:39	92	68	85.9	55.3	78.4	75.4	80.5	49.2	11.5	5.4	11.83	7.3	5.42	3.2
7–23	3	2–40	98.5	41.5	6.3	14:38	93	71	85.4	56.8	78.8	72.3	79.9	47.4	13.1	5.5	9.92	5.8	15.27	9.0
7–24	4	2–40	148.2	84.5	32.5	14:36	97	73	84.9	62.2	79.6	71.6	79.9	50.2	17.1	5.0	6.33	3.7	23.90	14.1
7–25	5	2–40	191.0	110.2	52.3	14:34	99	75	86.9	53.4	..	..	80.4	46.1	18.6	6.5	..	0	23.98	14.3
7–26	6	2–40	145.9	76.4	23.5	14:32	96	76	88.1	53.5	..	..	80.6	44.8	15.4	7.5	..	0	17.15	10.2
7–27	7	2–40	92.8	39.3	8.0	10:31	93	72	84.9	54.2	79.1	71.0	79.5	50.3	13.5	5.4	7.00	4.0	15.80	9.5
7–28	8	2–40	161.0	92.5	38.8	14:28	97	73	88.3	60.5	78.9	71.8	79.9	50.7	17.1	8.4	8.00	4.5	17.47	10.4
7–29	9	5–40	161.2	96.0	43.3	14:26	98	75	84.5	59.9	79.9	71.1	82.3	34.5	15.7	2.2	6.50	3.7	16.38	9.1
7–30	10	5–40	124.2	68.8	27.0	11:31	97	72	88.0	52.8	..	..	82.6	32.9	14.4	5.4	..	0	10.50	6.0
7–31	11	3–40	54.5	16.5	0	14:22	90	71	85.0	48.5	79.7	70.6	79.8	48.6	10.2	5.2	11.30	6.5	11.00	6.4
8–3	12	3–40	138.5	77.1	30.7	14:16	97	65	81.2	56.3	75.4	57.3	79.1	54.2	17.9	2.1	12.00	7.0	10.13	5.9
8–4	13	3–40	36.5	3.0	0	10:58	89	72	81.5	62.8	..	..	79.4	53.2	9.6	2.1	..	0	11.50	6.8
8–5	14	3–40	65.0	27.0	8.3	10:21	94	71	82.9	58.5	77.8	77.7	79.6	56.1	14.4	3.3	9.50	5.6	2.43	1.5
8–13	15	3–40	84.3	35.5	6.8	8:00	93	71	84.5	62.2	76.8	70.5	79.3	55.0	13.7	5.2	10.50	6.3	3.05	1.8
8–14	16	3–40	73.5	37.8	10.5	9:12	96	69	79.7	72.8	77.2	69.3	79.2	54.0	16.8	0.5	9.75	5.8	2.80	1.8
8–15	17	3–40	65.0	26.0	5.0	13:50	92	69	81.4	55.0	76.5	69.6	79.3	51.9	12.7	2.1	11.08	6.4	2.25	1.5
8–16	18	3–40	111.0	54.3	16.5	13:32	96	70	86.8	51.3	77.4	79.8	79.4	51.8	16.6	7.4	11.50	6.8	5.50	3.3
8–17	19	3–40	110.0	55.8	20.5	11:59	96	71	87.3	47.0	78.1	66.0	79.6	51.0	16.4	7.7	8.50	5.2	8.42	4.9
8–29	20	4–40	2.5	0	0	10:48	82	67	80.3	53.3	..	..	77.0	53.6	5.0	3.3	..	0	1.49	0.6

TABLE 2—COMPLETE DATA AND RESULTS FOR TESTS—1940 (*Continued*)

DATE 1940	TEST No.	SERIES No.	OVERALL TEST PERIOD			DEHUMIDIFIER AND DRY AIR COOLER OPERATION						RE-SATURATOR OPERATION, HOURS	TOTAL MOISTURE REMOVED LB.	HEAT ABSORBED BY COOLING PLANT DURING OVERALL TEST PERIOD BTU.			RATIO MOISTURE TO TOTAL LOAD
			Start of Cooling with Dehydration Unit to Start of Night Air Cooling			Time in Hours	Power Input Kwhr.	Gas Input Cu. Ft.	Cooling Water					Load Due to Moisture	Sensible	Total	
									Quantity in Gallons	Temperature							
			Start	End	Duration Hours					Inlet F	Rise F						
1	2	3	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36
7–21	1	2–40	10:12a	7:50p	9.63	6.68	4.7	630	1899	75.1	14.8	4.66	94.5	104,600	72,800	177,400	0.590
7–22	2	2–40	8:26a	8:41p	12.25	5.42	3.8	530	1439	74.8	17.0	0	114.4	125,600	16,500	142,100	0.884
7–23	3	2–40	7:08a	11:40p	16.53	15.27	10.9	1465	4602	75.1	14.6	12.60	173.9	192,900	192,800	385,700	0.500
7–24	4	2–40	6:00a	6:00a	24.00	23.90	16.7	2405	7035	75.0	15.3	23.90	241.4	271,300	336,700	608,000	0.446
7–25	5	2–40	6:00a	6,00a	24.00	23.98	16.5	2280	7215	74.9	15.1	23.98	227.8	254,800	362,700	617,500	0.413
7–26	6	2–40	6:00a	11:10p	17.17	17.15	11.7	1650	5180	75.0	15.0	17.15	166.3	185,500	266,800	452,300	0.410
7–27	7	2–40	6:10a	10:00p	15.83	15.80	11.0	1540	4691	75.2	15.1	15.80	151.7	170,400	221,100	391,500	0.435
7–28	8	2–40	6:00a	11:30p	17.50	17.47	12.2	1660	5229	75.5	15.3	17.47	173.0	192,300	261,200	453,500	0.424
7–29	9	5–40	6:00a	6:00a	24.00	16.38	10.5	1530	4975	75.2	13.9	0	298.1	328,600	75,400	404,000	0.813
7–30	10	5–40	6:00a	6:42p	12.70	10.50	6.9	970	3138	75.1	15.0	4.90	140.0	156,200	119,500	275,700	0.567
7–31	11	3–40	6:00a	7:27p	13.45	11.00	7.6	1050	3387	75.2	14.1	11.00	94.6	106,200	165,500	271,700	0.391
8–3	12	3–40	:15p	7:00a	14.75	10.13	7.0	990	3014	75.4	14.9	9.30	83.7	93,500	135,300	228,800	0.409
8–4	13	3–40	7:00a	9:30p	14.50	11.50	7.8	1120	3501	75.2	14.9	9.83	148.6	164,300	143,900	308,200	0.533
8–5	14	3–40	2:00p	5:55p	3.92	2.43	1.8	260	774	75.3	14.1	1.83	37.5	41,500	25,400	66,900	0.620
8–13	15	3–40	4:15p	9:15p	5.00	3.05	2.1	310	890	75.1	16.2	1.83	50.4	55,600	25,900	81,500	0.682
8–14	16	3–40	2:30p	7:55p	5.42	2.80	2.0	290	822	75.2	14.5	1.08	44.3	48,800	15,300	64,100	0.761
8–15	17	3–40	4:45p	7:30p	2.75	2.25	1.7	220	675	75.6	16.0	1.78	30.9	34,400	24,400	58,800	0.585
8–16	18	3–40	3:00p	10:30p	7.50	5.50	3.8	540	1649	75.2	15.2	5.00	58.6	65,300	75,600	140,900	0.463
8–17	19	3–40	12:45p	10:00p	9.25	8.42	5.9	810	2482	75.2	15.3	8.42	67.4	75,100	127,300	202,400	0.371
8–29	20	4–40	2:30p	5:10p	2.67	1.49	1.0	140	444	75.2	14.0	0	31.0	33,800	0	33,800	1.000

TABLE 3—TYPICAL OPERATING DATA AND RESULTS

Item	1	2	3
1. Date	8–14–40	7–25–40	7–29–40
2. Test Number	16	5	9
3. Series	3–40	2–40	5–40
4. Time of Day	6:10 p	2:00 p	2:00 p
5. Outdoor Air			
Dry Bulb, deg. F	76.0	97.0	98.0
Wet Bulb, deg. F	71.7	74.9	78.5
Dew Point, deg. F	69.9	65.6	71.3
Relative Humidity, per cent	81.0	34.5	41.0
Specific Humidity, grains per pound dry air	109.8	94.3	115.5
6. Indoor Air, average breathing level temperature			
1st story, deg F	78.0	80.0	81.4
2nd story, deg F	79.4	81.8	82.0
7. Indoor Air average at breathing level			
Dry Bulb, deg F	78.7	80.9	81.7
Wet Bulb, deg. F	65.0	66.7	59.9
Dew Point, deg. F	57.4	59.2	44.1
Relative Humidity, per cent	47.3	46.8	25.8
Specific Humidity, grains per pound dry air	70.0	74.9	42.2
8. Indoor Air entering return grille			
Dry Bulb, deg. F	77.5	79.1	80.3
Wet Bulb, deg. F	65.0	65.1	60.2
Dew Point, deg. F	58.1	57.3	45.8
Relative Humidity, per cent	50.5	46.5	29.0
Specific Humidity, grains per pound dry air	72.0	69.8	45.3
9. Ventilating Air			
Dry Bulb, deg. F	79.0	96.1	95.5
Wet Bulb, deg. F	72.9	76.6	75.4
Dew Point, deg. F	70.4	68.9	67.2
Relative Humidity, per cent	74.7	40.0	38.5
Specific Humidity, grains per pound dry air	111.8	106.1	99.8
10. Mixed Air, entering Dehumidifier			
Dry Bulb, deg. F	78.5	83.3	84.6
Wet Bulb, deg. F	67.6	68.2	64.5
Dew Point, deg F	62.2	60.5	52.3
Relative Humidity, per cent	56.5	45.0	32.0
Specific Humidity, grains per pound dry air	83.4	78.4	58.1
11. Mixed Air, leaving dehumidifier and entering dry air cooler			
Dry Bulb, deg. F	120.0	129.4	122.8
Wet Bulb, deg. F	72.0	73.0	69.3
Dew Point, deg. F	43.0	36.2	27.8
Relative Humidity, per cent	7.2	4.1	3.5
Specific Humidity, grains per pound dry air	40.8	31.2	21.7
12. Mixed Air, leaving Dry Air Cooler and entering Resaturation Cooler			
Dry Bulb, deg. F	78.0	77.6	78.1
Wet Bulb, deg. F	59.2	56.8	54.1
Dew Point, deg. F	45.4	39.3	30.0
Relative Humidity, per cent	30.8	24.8	16.5
Specific Humidity, grains per pound dry air	44.6	35.4	24.0
13. Mixed Air, leaving Resaturation Cooler			
Dry Bulb, deg. F	77.5	63.6	77 4
Wet Bulb, deg F	59.4	57.8	54.8
Dew Point, deg. F	46.4	54.1	33.3
Relative Humidity, per cent	32.8	70.5	19.5
Specific Humidity, grains per pound dry air	46.6	62.0	27.8

TABLE 3—TYPICAL OPERATING DATA AND RESULTS—*Continued*

Item—*Cont.*	1	2	3
14. Air Temperature drop through Cooling Unit, deg. F	1.0	19.7	7.2
15. Temperature cooled air leaving registers, average deg. F	78.3	71.4	79.5
16. Air Temperature rise in ducts and casing, deg. F.	0.8	7.8	2.1
17. Basement Air Temperature at breathing level, deg. F	78.0	79.3	81.1
18. Quantity of Air through Cooling Unit			
Cubic feet per minute	804	804	804
Pounds per hour	3622	3622	3622
Density of air, lb. per cu. ft	0.075	0.075	0.075
19. Number of house air recirculations per hr	3.4	3.4	3.4
20. Ventilating air			
Cubic feet per minute	232	232	232
Pounds per hour	1041	1041	1041
Density of air, lb. per cu. ft	0.075	0.075	0.075
21. Moisture removed from air, pounds per hr	19.11	8.51	15.67
22. Heat given up by air; total B.t.u. per hour	21,810	26,770	23,510
Heat in moisture removed from air, B.t.u. per hour	20,920	9,340	17,200
Heat given up by moisture remaining in air, B.t.u. per hr	20	330	40
Heat due to moisture in air, total B.t.u. per hr.	20,940	9,670	17,240
Per cent of total	96.0	36.1	73.3
Sensible heat			
B.t.u. per hr	870	17,100	6,270
Per cent of total	4.0	63.9	26.7
23. Fan Speed, r.p.m.	510	510	510
24. Fan Motor Data			
Size h.p.	0.75	0.75	0.75
Measured power rate, Kw	0.64	0.60	0.56
25. Cooling Unit Data			
Measured power rate, Kw	0.71	0.69	0.64
Measured Gas rate, cu. ft. per hour	103.6	9.51	9.34
Heating Value of Gas, B.t.u. per cu. ft	1000	1000	1000
26. Activation Air entering house			
Dry Bulb, deg. F	76.0	98.1	97.3
Wet Bulb, deg. F	72.3	76.1	75.8
Dew Point, deg. F	71.0	67.5	67.3
Relative Humidity, per cent	84.0	37.0	38.0
Specific Humidity, grains per pound dry air	114.0	100.5	100.0
27. Activation Air leaving house			
Dry Bulb, deg. F	147.0	160.0	159.7
Wet Bulb, deg. F	99.0	99.2	96.6
Dew Point, deg. F	89.0	85.8	81.4
Relative Humidity, per cent	19.5	13.0	11.3
Specific Humidity, grains per pound dry air	210.0	189.0	164.0
28. Water Temperature in dry air cooler			
Inlet, deg. F	75.0	74.9	75.0
Outlet, deg. F	89.0	90.2	88.8
Rise, deg. F	14.0	15.3	13.8
29. Quantity of cooling water			
Pounds per hour	2465	2540	2545
Gallons per hour	296	305	305
30. Heat absorbed by cooling water, B.t.u. per hour.	34,510	38,860	35,120

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rise in temperature of the air in passing through the furnace casing and ducts was 0.8 F. Of the total amount of heat removed from the air (Item 22), only 4 per cent represented the heat absorbed in reducing the dry-bulb temperature and 96 per cent represented the heat absorbed in removing the water vapor.

As shown in Fig. 6, outdoor air had been circulated through all the rooms of the house from 9:15 p.m. to 7:00 a.m. during the night previous to the test.

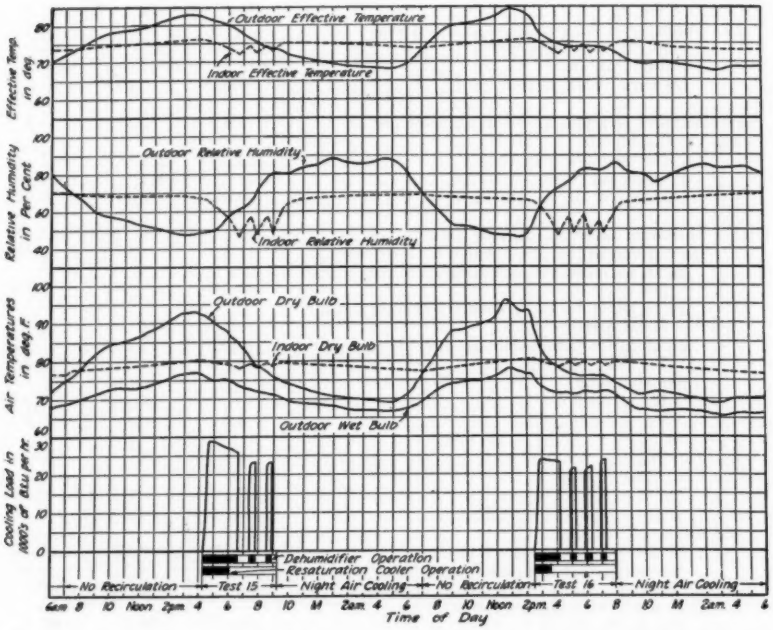
Outdoor Effective Temperature
Indoor Effective Temperature
Effective Temp. in deg.
Outdoor Relative Humidity
Indoor Relative Humidity
Relative Humidity in Per Cent
Outdoor Dry Bulb
Indoor Dry Bulb
Outdoor Wet Bulb
Air Temperatures in deg. F.
Cooling Load in 1000's of Btu per hr.
Dehumidifier Operation
Resaturation Cooler Operation
No Recirculation
Test 15
Night Air Cooling
No Recirculation
Test 16
Night Air Cooling
6am 8 10 Noon 2pm 4 6 8 10 M 2am 4 6 8 10 Noon 2pm 4 6 8 10 M 2am 4 6
Time of Day

FIG. 6. GRAPHIC LOG OF AIR TEMPERATURES, RELATIVE HUMIDITIES, AND COOLING LOAD—TESTS 15 AND 16, SERIES 3-40, AUG. 13 AND 14, 1940

At 7 a.m. the windows on the second story were closed and the basement fan was stopped. By 2:30 p.m. the temperature on the second story had increased to 81 F and the cooling unit was started. During the period from 2:30 p.m. to 7:50 p.m. the cooling plant operated intermittently to maintain the dry-bulb temperature on the second story at about 80 F and the relative humidity at about 50 per cent. At 7:50 p.m., when the outdoor effective temperature became lower than the indoor effective temperature, the windows on the second story were opened and the cooling with outdoor air at night was begun.

With the types of cooling plants used in previous years, on days during which the plant operated intermittently, a period at the start of the test and the off-periods were accompanied by relatively high humidities which were not conducive to comfort. With the dehydration cooling unit used in 1940,

the rise in relative humidity during the off-periods was limited by means of the operation of the dehumidifier, and the comfort conditions during the off-periods were entirely satisfactory. At the beginning of each test, however, the high relative humidities and effective temperatures existing prior to and immediately following the start were accompanied by comfort conditions that were not entirely satisfactory during those periods. As shown by Fig. 6, introduction of cool humid air during the night resulted in an indoor relative humidity of 69.5 per cent at 7 a.m. Prior to the start of the cooling unit the relative humidity decreased from 69.5 per cent to 65.5 per cent, and the effective temperature rose from 73.6 deg at 7 a.m. to 76.1 deg at 2:30 p.m. while the dry-bulb temperature rose from 77.3 F to 80.4 F. Thus, for a period of about four hours previous to and ½ hour subsequent to the start of the test the indoor effective temperature exceeded 75 deg which has been considered as the upper border line for comfort.

Constant Effective Temperature. Supplemented by Cooling with Outdoor Air at Night. (*Series 2-40.*) The initial period of effective temperatures exceeding the upper limit of comfort, occasioned by delaying the starting of the cooling unit until the dry-bulb temperature on the second story rose to 81 F, could be eliminated by starting the plant earlier, thus bringing the dehumidifier and dry air cooler into operation, when the effective temperature rose to 74 deg. Thereafter, the cooling unit was operated with the thermostat set to maintain 80 F and the humidistat set to maintain 50 per cent relative humidity. The series of tests made under these conditions has been designated as *Series 2-40.* The results obtained on two typical hot days are shown in Fig. 7. The test made on July 24, 1940, was preceded by supplementary cooling with outdoor air at night. However, owing to the fact that during the night of July 24 the outdoor effective temperature never dropped below 74 deg, cooling was continued without opening the windows, and the test of July 25 was not accompanied by supplementary cooling with outdoor air on the preceding night. The data shown in the second column of Table 3 were obtained on July 25, 1940, at 2:00 p.m. at which time the dehumidifier, the dry air cooler, and the resaturation cooler were all operating. The heat represented by the extraction of water vapor was in this case 36.1 per cent of the total amount of heat removed from the air. (Item 22.)

As shown in Fig. 7 the cooling unit operated continuously over a period beginning at 6:00 a.m. on July 24, 1940, and extending beyond 6:00 a.m. on July 26. During the entire test period the effective temperature was maintained between 72.5 deg and 74.5 deg and the comfort conditions were entirely satisfactory. On milder days than those shown in Fig. 7 the cooling plant operated intermittently with equally satisfactory results. The increase in comfort resulting from starting the cooling unit when the effective temperature rose to 74 deg, instead of starting the cooling unit when the dry-bulb temperature on the second story rose to 81 F, was accompanied by an increase in the total time of operation per day.

Constant Effective Temperature. Cooling Not Supplemented by Outdoor Air at Night. (*Series 4-40.*) Owing to the fact that the advantage of supplementary cooling with outdoor air at night is sometimes discounted, and the continuous use of air conditioning without the admission at any time of outdoor air in excess of ventilation requirements is advocated, a series of tests under the latter conditions was run. It may be noted that by operating in

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this manner (*Series 4-40*), certain periods were covered during which outdoor air at night would have been feasible, while on other warm nights the outdoor effective temperature did not become less than that indoors before midnight, and no air from outdoors would normally have been introduced. In the latter case *Series 2-40* and *3-40* would have merged into *Series 4-40* as illustrated by the test made on July 25, 1940, and shown in the right hand portion of Fig. 7.

Tests included in *Series 4-40* were made with the windows on the first and second stories in the Residence closed day and night, irrespective of outdoor

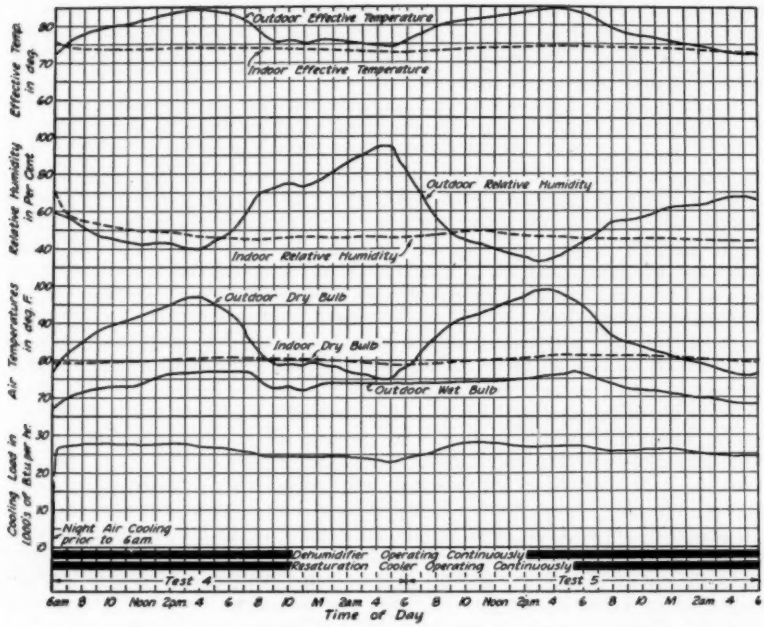
Outdoor Effective Temperature
Indoor Effective Temperature
Effective Temp in deg
Outdoor Relative Humidity
Indoor Relative Humidity
Relative Humidity in Per Cent
Outdoor Dry Bulb
Indoor Dry Bulb
Outdoor Wet Bulb
Air Temperatures in deg F.
Cooling Load in 1000's of Btu per hr.
Night Air Cooling prior to 6 am.
Dehumidifier Operating Continuously
Resaturation Cooler Operating Continuously
Test 4
Test 5
6am 8 10 Noon 2pm 4 6 8 10 M 2am 4 6 8 10 Noon 2pm 4 6 8 10 M 2am 4 6
Time of Day

FIG. 7. GRAPHIC LOG OF AIR TEMPERATURES, RELATIVE HUMIDITIES AND COOLING LOAD—TESTS 4 AND 5, SERIES 2-40, JULY 24 AND 25, 1940

temperatures, and with the ventilating fan operating continuously to supply outdoor air equivalent to one air change per hour. The indoor effective temperature was automatically maintained at approximately 74 deg. Occasionally, in very mild weather, operation of the dehumidifier and dry air cooler alone resulted in an effective temperature approaching 72 deg even though the relative humidity did not drop materially below 50 per cent. In these cases the plant was stopped manually for short periods of time. The comfort conditions were satisfactory, and the variations in relative humidities were not large.

On warm, humid nights the contrast between the indoor and outdoor air conditions is such that there would be no objection to keeping the windows

closed and depending upon the cooling unit to maintain the desired indoor temperature and humidity. On such nights the advantage of this method of operation is unquestionable. Furthermore, from the tests made with cooling during the day supplemented by cooling with outdoor air at night, as shown in Fig. 6 and in the left hand portion of Fig. 7, it is evident that the introduction of outdoor air into the house at night always resulted in a high indoor humidity. After the house was closed in the morning and the cooling plant was started, the dehumidifier operated at full load for a considerable time before the indoor humidity was brought back to normal at about 50 per cent. Hence, on comparatively warm nights, the advantage effected by the introduction of outdoor air at night, as measured by the reduction of the indoor dry-bulb temperature, was offset by the rise in relative humidity.

However, many nights occur during which the outdoor temperature is sufficiently lower than the indoor temperature, so that, if an effective means of cooling with outdoor air at night is available, the advantage represented by lowering the indoor temperature more than offsets the disadvantage resulting from the increase in indoor humidity. Under these conditions cooling by means of outdoor air at night is justifiable, even though such supplementary cooling involves the labor of opening and closing the windows each day. On such nights, when the effective temperature outdoors is less than that indoors, most home-owners would be strongly impelled to open the windows, and any consideration of an ideal, convenient method of cooling the house will have to take into account the fundamental aversion to being confined indoors when the outdoor conditions are at least equally satisfactory. It is true that by the admission of very large quantities of ventilating air the desire to open the windows might be minimized. However, such quantities would have to be much greater than the one air change per hour required to minimize odors.

Low Relative Humidity. (*Series 5-40.*) As shown in Fig. 8, from 6 a.m. on July 29, 1940, to 6:42 p.m. on July 30, 1940, the plant was operated with the controls set to maintain approximately 83 F and 30 per cent relative humidity. At 6:00 a.m. on July 29, cooling with outdoor air at night was ended and cooling by means of the dehydration unit was begun.

The data shown in the third column of Table 3 were obtained after the dehumidifier and dry air cooler had been operating continuously for 8 hours. From Item 22 it may be observed that of the total amount of heat removed from the air, 26.7 per cent represented the heat absorbed in reducing the dry-bulb temperature and 73.3 per cent represented the heat absorbed in removing the water vapor.

It required practically nine hours of continuous operation of the dehumidifier and dry air cooler in order to reduce the relative humidity from 71.8 per cent at 6 a.m. to 25.5 per cent at 2:55 p.m., or a total reduction of 46 per cent. The reduction in humidity was extremely rapid during the first two hours of plant operation and much slower thereafter. After a period of cooling with outdoor air at night, the moisture which was removed consisted not only of that removed from the ventilating air and that due to occupants, but also of the moisture abstracted from the plaster, woodwork, and furnishings of the Residence.

Without considering the merits or demerits from the standpoint of comfort of operating with a low humidity and a comparatively high dry-bulb temperature, it is evident that if, for any reason, this method of operation is to be

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used, it cannot be used to any advantage in connection with supplementary cooling with outdoor air at night.

Cooling Loads and Effectivenes of Insulation. The calculated cooling loads, consisting of the sensible heat, moisture, and total heat loads, for the insulated and uninsulated structure are presented in Table 4. The application of insulation to the sidewalls and ceiling probably would not change the amount of water vapor brought in by the ventilation air or that given off by the occupants. Hence, the moisture load as shown in the second column of the table remained constant for the two cases. Any saving effected by the application

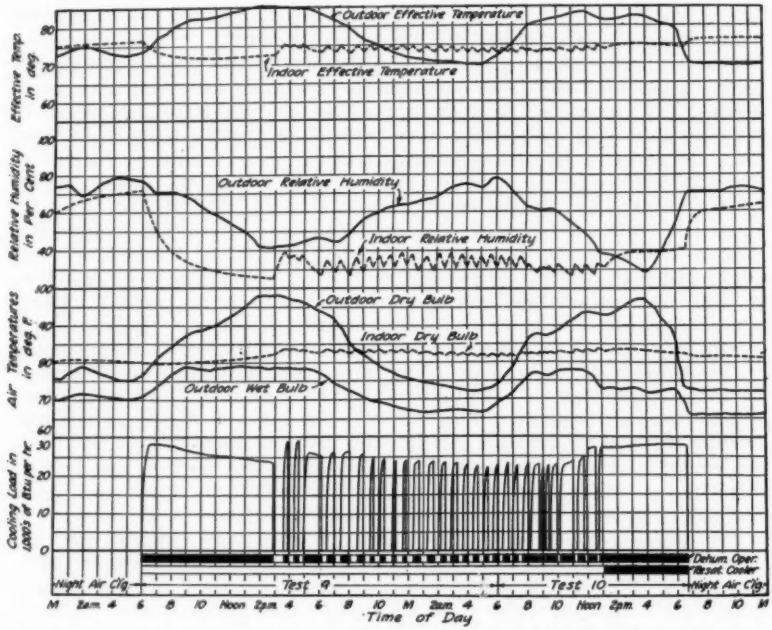
Outdoor Effective Temperature
Indoor Effective Temperature
Effective Temp. in deg.
Outdoor Relative Humidity
Indoor Relative Humidity
Relative Humidity in Per Cent
Outdoor Dry Bulb
Indoor Dry Bulb
Outdoor Wet Bulb
Air Temperatures in deg F
Cooling Load in 1000's of Btu per hr
Night Air Clg
Test 9
Test 10
Dehum. Oper. Resat. Cooler
Night Air Clg
M 2am 4 6 8 10 Noon 2pm 4 6 8 10 M 2am 4 6 8 10 Noon 2pm 4 6 8 10 M
Time of Day

FIG. 8. GRAPHIC LOG OF AIR TEMPERATURES, RELATIVE HUMIDITIES, AND COOLING LOAD—TESTS 9 AND 10, SERIES 5-40, JULY 29 AND 30, 1940

of insulation, however, would be reflected in a reduction in the sensible heat load. As shown by the first column in Table 4, the calculated sensible heat load was reduced from 32,260 Btu per hour to 20,060 Btu per hour representing a saving of 12,200 Btu per hour, or 37.8 per cent. The assumption is usually made that the reduction in the calculated hourly cooling load is indicative of the reductions to be expected in the total cooling load and operating costs for a given day or for a given season.

The sensible heat load obtained by test, expressed as the British thermal units absorbed per day in reducing the dry-bulb temperature of the air, is shown in Fig. 9 plotted against the degree-hours above 85 F per day. *Curve A* represents the results obtained in the insulated structure from the tests,

TABLE 4—CALCULATED COOLING LOAD FOR INSULATED AND UNINSULATED RESIDENCE

	CALCULATED COOLING LOAD BTU PER HOUR		
	Sensible	Latent	Total
Uninsulated	32,260	5,240	37,500
Insulated	20,060	5,240	25,300
Reduction	12,200	0	12,200
Reduction in per cent	37.8%	0	32.6%

NOTE: The calculated cooling load was based on accepted method given in ASHVE GUIDE. Conditions assumed were: Outdoor dry bulb = 95 F; Wet bulb = 76 F; Indoor dry bulb = 80 F; Relative humidity = 50 per cent; Ventilation air = one change per hour.

designated as *Series 3-40,* which were started when the dry-bulb temperature on the second story rose to 81 F. *Curve B,* which has been reproduced from Engineering Experiment Station Bulletin 290, shows results from similar tests (*Series 3-34*) made in the uninsulated structure. A comparison of the *Curves A* and *B* shows that over a wide range of weather conditions, the average reduction in sensible heat load due to the insulation was approximately 33 per cent. This value is probably conservative due to the fact that more heat was lost to the basement from the dehydration unit than from the condensing unit used in 1934, and the basement temperature was accordingly about 2 F higher during the 1940 tests. It also agrees reasonably well with the calculated reduction in sensible heat load of 37.8 per cent given in the first column in Table 4. Corresponding percentage reductions in the total cooling load, which includes both moisture and sensible loads, would be smaller than the reduction in sensible heat load alone, as illustrated by a comparison of the values in the first and third columns in Table 4. The operating costs of the plant would be dependent upon the total cooling load, and not upon the sensible heat load alone. The moisture load is extremely variable but on the assumption that it was approximately the same for the seasons of 1934 and 1940 the reduction in total load effected by the insulation was of the order of 25 per cent.

It may be observed from the first column in Table 4 that the moisture load, as calculated in standard practice from that brought in by infiltration, or the ventilating air, and that given off by occupants was 5240 Btu per hour. The actual hourly moisture loads, obtained after periods of at least six hours of continuous operation, on three days on which the outdoor dry-bulb temperature was approximately 95 F and the wet-bulb approximately 76 F, corresponding to design conditions, were 9640, 9860, 9860, 10610, 10750, 11440 and 11620 Btu per hour. This large discrepancy probably results from the fact that water vapor is not merely brought into a structure as an accompaniment of air leaking in, but permeates all materials and enters all openings in amounts that are dependent on the difference in vapor pressure, and which may be independent of the amount of air inleakage. In the light of this fact it is probable that some revision of the commonly accepted practice in the calculation of moisture loads should be made.

The sensible heat load for the tests made with constant effective tempera-

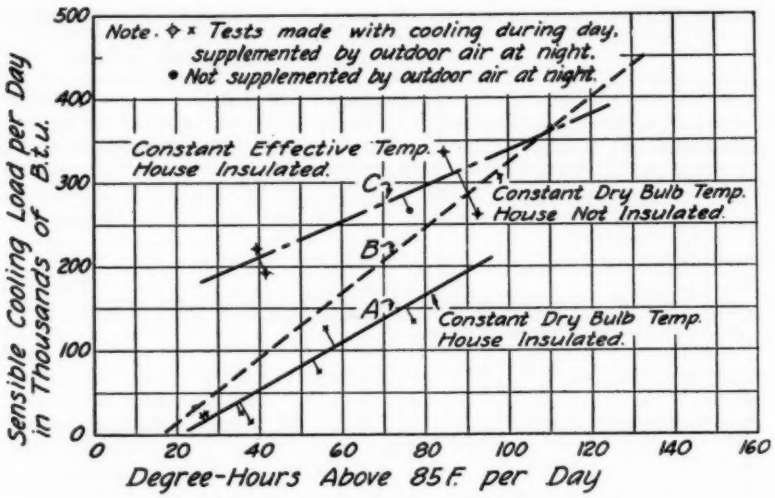

FIG. 9. SENSIBLE COOLING LOAD PER DAY

ture maintained in the Residence (*Series 2-40*) is designated as *Curve C* in Fig. 9. It may be observed by comparing *Curve C* with *Curve A* that the improvement in comfort conditions, resulting from maintaining a constant effective temperature of 74 deg instead of a constant dry-bulb temperature of 80 F, was obtained at the expense of a considerable increase in the total time of plant operation and hence in operating cost. It is probable that for extremely hot days, represented by larger values of degree-hours, *Curves A* and *C* would tend to converge.

SUMMARY AND CONCLUSIONS

The following summary and conclusions may be considered as applying to the Research Residence and the conditions under which the tests were conducted.

1. The method of operation by which the cooling plant was started when the indoor effective temperature rose to 74 deg after a period of supplementary cooling with outdoor air at night, was more satisfactory than the one by which the cooling plant was started when the indoor dry-bulb temperature rose to 81 F, in that it eliminated the period of discomfort occurring just previous to the start of the cooling unit.

2. By the use of the dehydrating unit it was possible to maintain comparatively uniform relative humidity, and consequently uniform comfort conditions, irrespective as to whether the plant operated continuously or intermittently.

3. The use of insulation in the side-walls and ceiling effected a saving of approximately 33 per cent in the sensible heat load and of the order of 25 per cent in the total heat load.

4. Lack of agreement between the calculated design moisture load and the actual moisture load indicates that moisture permeating the structure may be independent of the amount of infiltration or ventilation air, and suggests that some revision may be advisable in the commonly accepted method for calculating moisture load.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with the summer cooling investigation (1940) in the Research Residence[7] at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is due M. K. Fahnestock, Research Associate Professor, for his interest and assistance in this work, and to H. R. Stover, Research Graduate Assistant, and J. C. Miles, Instructor, for services rendered in connection with the investigation. Asknowledgment is also due to the *American Gas Association* for active cooperation, and to the various manufacturers who furnished instruments and equipment.

DISCUSSION

C. F. BOESTER (WRITTEN): The authors are to be complimented on the thoroughness with which they made the investigations. In my opinion this is a great contribution, particularly from the viewpoint of vapor flow in the summer time from the outside to the inside of enclosures being conditioned.

Nowhere in any set of heat gain calculations has there been allowance made for substantial heat gains in the form of vapor flow. More often than not what otherwise would have been an unusually well-functioning summer cooling system, the job has turned out to be a poor one because of unaccounted for sources of heat through the flow of vapor. Many jobs also operate and are very satisfactory except under extreme conditions when they limp quite badly due to rapid heat gain from vapor flow.

We have been making rather elaborate studies in vapor flow both in the winter time from the inside of an enclosed space to the outside, and in the summer time from the outside to the inside, and we have almost come to the conclusion that it is more important to make an investment in a good vapor barrier than in insulation as a means of reducing the rate of heat exchange in an enclosed space, in an endeavor to maintain comfort conditions.

It has been established many times that insulation in itself pays dividends and I would take no quarrel with such findings. I would only say that a vapor barrier pays greater dividends. Some worthy studies are now being made of various types and kinds of vapor barriers and many new such vapor barriers are being developed, but certainly further investigation should still be made.

As to the other parts of the investigation, they are, of course, likewise worthwhile; but in my opinion concern the commercial aspect of air conditioning in that they contribute data pertaining to the merits of one method of dehumidifying the air as opposed to another.

H. F. HUTZEL (WRITTEN): In the summary and conclusion the authors call attention to lack of agreement between actual and calculated moisture loads and suggest that some revision may be advisable in the accepted method of calculated moisture load. I do not believe that the accepted method of calculating latent loads is incorrect but do believe that the latent load, on basis on which these tests were conducted, was incorrectly calculated.

In Table 4 the calculated hourly latent load is given as 5,240 Btu per hour. This is the correct latent load contributed by 14,170 cu ft of air per hour from dew-

[7] The Research Residence in Urbana, Illinois, was built, furnished and completely equipped specifically for research work in warm air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

point corresponding to outside design conditions of 95 F dry-bulb 76 F wet-bulb to dew-point corresponding to inside design conditions of 80 F dry-bulb 67 F wet-bulb. This latent load calculation does not include latent load contributed by increased volume of outside air circulated during the night at which time the air conditioning equipment was not in operation.

Outside air temperatures and per cent relative humidities recorded in Fig. 7 show that the dew-point of outside air during the night is approximately the same as during the daytime. This was approximately 70 F. Since the inside dew-point during these tests was approximately 58 F it is logical to expect that when this night air at 70 F dew-point is circulated through the residence, considerable moisture will be absorbed by the building materials and furnishings to be given up in turn when the air conditioning equipment is operated in the daytime. Had the air conditioning system been operated continuously day and night, without increased volume of outside air circulated during the night, the actual latent load would have been more closely in accord with calculated latent load. Whereas the calculated latent load in Table 4 is 20.7 per cent of the total load, it should be pointed out that this ratio applies at design conditions only. During the night operation, with decreased sensible load, this percentage of latent to total heat ratio increases and over a 24-hour period the average would be higher.

F. G. SEDGWICK[8] (WRITTEN): This paper is an exceptionally complete and able discussion of a very interesting series of tests conducted in the Research Residence at Urbana, Ill., during the summer of 1940.

The *National Warm Air Heating and Air Conditioning Association* takes this opportunity to express its sincere appreciation to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for this and other cooperative research programs which have been carried on through their cooperation with the research staff and research equipment at the University of Illinois.

Each research program in which the ASHVE have cooperated with the *NWAHACA*, has, in our opinion, resulted in material and valuable addition to the store of knowledge relating to the subject of research.

Referring particularly to the paper under discussion, this paper covers a research project on a gas-fired dehydration cooling unit.

This research program was carried on in our Research Residence, which had been insulated since the last previous cooling program and, therefore, there was more of interest in the results than the results of the tests on the gas-fired dehydration unit itself.

Of particular interest to many was the 33 per cent saving in the sensible heat load, resulting from the use of insulation in the side walls and ceiling, and also the fact that our attention was so definitely called to the fact that this saving is not a saving in the total heat load which saving was of the order of 25 per cent, due, of course, to the fact that the latent heat load did not materially change.

I was very much interested in the suggestion for additional study leading to the possible revision in the commonly accepted method of calculating moisture load, and in the suggestion that possibly moisture permeating the structure may be independent of the amount of infiltration or ventilating air.

It was further interesting to note the exceptionally satisfactory humidity control made possible through the use of the dehydration unit.

May I call the attention of the Society to the fact that research results obtained in the Research Residence even though they may not contribute new knowledge to the art (and many of them do), result in a dramatization and popularization and public acceptance, which are gained with great difficulty through other less well known research channels.

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[8] Vice-President, Waterman-Waterbury Co., Minneapolis, Minn. Chairman, Research Advisory Committee, *National Warm Air Heating and Air Conditioning Association.*

The *National Warm Air Heating and Air Conditioning Association* welcomes the use of its equipment for cooperative research on projects approved by its Research Advisory Committee, and takes advantage of this opportunity to invite the ASHVE to continue to study the problems of the industry through our facilities on any mutually desirable, cooperative program.

J. DONALD KROEKER (WRITTEN): The Pacific Northwest, being an area which has consistently cool nights during the warm summer weather, we are very much interested in the paper on summer cooling in the Research Residence.

An interesting indication is that night cooling in the Research Residence apparently resulted in absorption by hygroscopic materials within the residence of sufficient moisture to appreciably affect the amount of latent cooling required. If this indication is the logical explanation for the increase in required heat absorption, then it appears some advantage might obtain from using higher indoor relative humidities since the amount of mosture absorbed by hygroscopic materials within the residence must bear some direct relation to the difference between the maintained indoor relative humidity and the relative humidity of air used for night cooling. With this in mind it seems that some appreciable economy in operation should result from maintained higher indoor relative humidity with simultaneously lower indoor dry-bulb temperatures, but still maintaining the same effective temperature.

Is information available on the effect on required heat absorption due to difference in relative humidities of air used for night cooling and on desirability of maintaining constant effective temperatures with higher relative humidities?

J. DEB. SHEPARD (WRITTEN): Careful consderation of the work done at the Research Residence is very important, for these reports are on procedures which may prove the answer to the development of a real domestic summer air conditioning market, which would obviously offer our industry greater opportunities than can be expected from either the commercial or industrial field.

The technical data set forth in this interesting paper amply justifies the work done on it by the research associates, the Society and other organizations supporting this work.

I would like, however, to place a little added emphasis upon the procedure followed in this project. I believe that this is the first time that the Society has had an extensive report upon a summer air conditioning project, which depends primarily upon pure dehumidification for its effectiveness.

One interesting feature of this application is worthy of special note. Cooling is provided, when necessary, by the re-evaporation of moisture; by lowering the temperature at the expense of raising the humidity—but when no cooling is required, the apparatus provides lower humidities than are normally called for, which is in striking contrast to conditions afforded by systems which depend primarily upon temperature reduction for dehumidification effect. A revelation of the low percentage of time when pure cooling is required emphasizes the importance of this inherent characteristic of the dehydration system.

It is interesting to note the authors' comment upon the disagreement between calculated and actual dehumidification loads at low humidities. Some interesting conclusions on this subject have been presented by commercial agencies working in the low humidity field. Anyone who has considered the problem of maintaining abnormally low humidities, as are required by certain industrial processes, fully appreciates the importance of the problem of moisture capacity of normal structures, and moisture permeation of common structural materials.

M. M. MCCLURE:[9] In connection with this extra load from humidity that is unaccounted for, I would like to relate an experience that was had in the Railway field in the investigation of air conditioning of railway cars. In Chicago, several

[9] School of Mechanical Engineering, Purdue University, Lafayette, Ind.

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intensive tests were made, in which there was a regain of moisture that was unaccounted for during the early period of progress of the test. However, after the testing had progressed for some time, it was found there was an infiltration of water vapor through the exhaust grill of the car. There was a splendid example of a counter-flow, air in one direction, and vapor in the other direction. The car was carefully sealed, except for the fresh air intake, and the exhaust. The added load for the fresh air was accounted for. Aside from this there was this extra infiltration of water vapor. The only place that this could be accounted for was the counter-flow in the exhaust grill, air leaving, water vapor entering.

MR. BOESTER: I happen to have read Mr. Hutzel's comment on this paper. I think his comments are interesting; that the drapes, plaster, etc., in the house may have absorbed an unreasonable amount of moisture during the night operation of the air cooling which was surrendered in the daytime during dehumidification operation.

N. W. DOWNES: It does not surprise me very much that there is a movement of water vapor in these directions, by infiltration, or through other holes in the building, or even from material in the building. If you will refer to the tabulation in Mr. Fleisher's paper,[10] you will notice they talk about grains per pound in the neighborhood of around 100, and in terms of per cubic foot, perhaps being in the neighborhood of 13 or 14.

Those correspond to the dew-points we ordinarily talk about in air conditioning, when we talk about comfort. But, if you put this other type of equipment in, it really dehydrates the air. You are talking about dew-points, which you do not ordinarily think of; grains of moisture per cubic foot at 37 deg are about two and a half.

There is a corresponding difference between the vapor pressure in the atmosphere on the outside of the building and the vapor pressure of the vapor within the building and of about the same ratio. Those numbers, give a very good idea of the differential pressure, as applied to water vapor, available to make it flow; some place in the neighborhood of 13 as compared to two and a half.

PROFESSOR KRATZ: Most of the comments seem to have centered around the question of the travel of moisture vapor, independent of the infiltration of air. This phenomenon has been observed under other conditions, particularly in gas heating, where we can very nearly, definitely prove that a travel of moisture from flue gases occurs out through the back draft diverter against the flow of air into the back draft diverter. The phenomenon is not at all different in this case.

Two or three comments have indicated that the commentors consider that we found this only in cases of low humidity. That is not true. We found this discrepancy between the calculated and actual moisture load at normal humidities of about 50 per cent. Mr. Hutzel's comment that this discrepancy might be attributed largely to storage in the building rather than to travel of moisture against the direction of air flow certainly merits consideration. However, Fig. 7 shows that, although the relative humidity was comparatively high when the cooling unit was started, it was reduced to 50 per cent after 6 hours of operation, in spite of a rising outdoor wet-bulb. From then on the humidity remained practically constant for a period of 42 hours with the machine running continuously. On this evidence it would seem that a period of 6 hours of running was sufficient to take care of any moisture stored in the structure after operation with outdoor air at night. Since all of the figures mentioned in the paper were obtained after at least 6 hours of continuous operation it seems probable that they are representative of conditions after stabilization had occurred.

[10] Comfort and Health and Temperature—A Mathematical Solution, by W. L. Fleisher and W. L. Fleisher, Jr. (ASHVE TRANSACTIONS, Vol. 47, 1941.)

No. 1178

A LABORATORY METHOD FOR CYCLIC HEAT MEASUREMENTS ON WALLS AND ROOFS

By E. R. QUEER* AND F. G. HECHLER,** STATE COLLEGE, PA.

THE guarded hot-box method for measuring the thermal resistance of wall or roof structures under steady state conditions, that is with constant temperatures on each side of the wall, has been well developed. If proper precautions are used this method gives satisfactory values not only for the over-all resistance of the structure but also yields results for the individual components of the wall agreeing closely with those obtained with the guarded hot plate. This paper describes the extension of the hot-box method of testing to the measurement of heat flow when one side of the wall is subjected to a varying temperature simulating outdoor conditions while the air on the other side is kept at a constant temperature. With the development of air conditioning this problem has become increasingly important because of the effect of heat capacity, of sunshine, and of normal daily cyclic temperature changes on the heat transmission of complete wall structures. The necessity for a suitable method for measuring cyclic heat flow has been frequently pointed out in the literature.[1-9]

This problem has been approached both analytically and experimentally. The analytical solution, based on an application of Fourier's series, is tedious and complex and beyond the ability of the average user. The experimental method depends on the use of a heat-flow meter, or its equivalent, to measure the flow through the wall or roof of the existing structure or of a special test house built for the purpose. The experimental method is laborious and expensive and is handicapped by the erratic and often unsuitable temperatures on one or both sides of the structure. The present method obviates many of these difficulties and makes it possible to carry out the required measurements under predetermined conditions economically and expeditiously.

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* Assistant Professor of Engineering Research, Engineering Experiment Station, The Pennsylvania State College. MEMBER of ASHVE.
** Director, Engineering Experiment Station, The Pennsylvania State College.
[1-9] Numbers refer to Bibliography.

Presented at the 47th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Kansas City, Mo., January, 1941.

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THE TEST EQUIPMENT

The test equipment, which is a modification of the guarded hot box, is shown in Fig. 1. The wall to be tested is installed so that the outside of the wall, which is to be subjected to the variable temperature corresponding to outdoor conditions, faces an open chamber (*A*) provided with means for heating and cooling. On the other side of the test wall is fitted a removable guarded hot box, one side of which is closed by a standard panel (*B*), through which the heat flow remains constant during a given test. The thermal resistance of this panel is chosen so that the heat flow through it for fixed air temperatures in the cold room (*C*) and in the guarded box (*D*) will always be greater than the heat flow through the wall under test. During a test the heat passing through the wall changes with the temperature in chamber (*A*); consequently

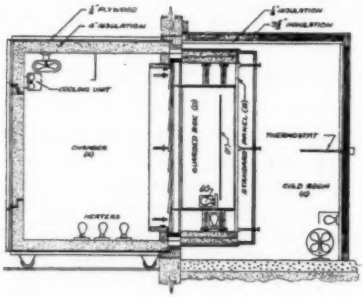

FIG. 1. MODIFIED GUARDED HOT-BOX ASSEMBLY WITH TEST WALL IN POSITION

in order to keep a constant rate of heat flow through panel (*B*) the heat supplied to the guarded box by the heaters (*E*) must be changed correspondingly.

The guarded box (*D*) has a light 1-in. wood frame covered on both sides with hard fiber board. The 1-in. space between the fiber boards is filled with a high grade mineral wool to provide some thermal insulation. The outer surfaces of the fiber boards are covered with 16-oz copper; the sheets on the two surfaces are electrically insulated from each other to facilitate the installation of thermocouples for obtaining thermal equilibrium between the inside and the outside of the box. The copper is painted black to minimize reflection of radiant heat. A black shield (*F*) 42 x 42 in. is placed 6 in. from the standard panel (*B*) to minimize radiation from the plaster surface to the standard panel which is at a lower temperature than the test wall.

The test wall consists of a common type of frame construction with a three coat, 5/8-in. thick plaster on 3/8-in. gypsum plaster base, 2 x 4-in. studs, 16 in. on centers, 25/32-in. wood sheathing, building paper, and 8-in. beveled siding. The construction and method of holding the wall together are shown in Fig. 2.

The wall is assembled with bolts and screws to make it easy to take apart to add or remove insulation. The method of assembly did not change the conductance from that normally expected for a wall of this type. The test area is 4 x 4 ft. in the center of the panel and the studs are arranged so that three full stud widths were covered by the guarded box (D). It should be noted that the air spaces within the wall covered by the test area are isolated from those covered by the guard ring.

The electrical circuits supplying heat to the component parts of the equipment along with the automatic control circuits are shown in Fig. 3. The

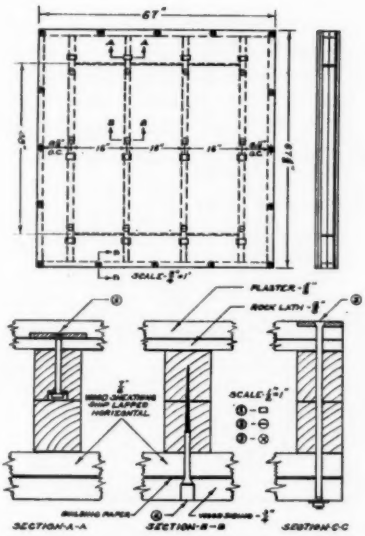

FIG. 2. CONSTRUCTION OF TEST WALL

temperature cycle on the outside of the wall was produced with exposed electric lights, fans, and a cooling unit. The heat flowing into or out of the wall on the outside was not measured. The heater units in the guarded box (D), and in the guard surrounding this box, consist of 120-w, 110-volt, clear, carbon filament electric bulbs. These are operated on low voltage and are carefully shielded to avoid direct radiation to the panel and meter. The heater units are located at the lowest level in the space being heated so as to make the temperature gradient between the bottom and top a minimum. The heat distribution was dependent on natural convection, as no fans were used on the plaster side of the panel. Shielded electric bulbs are also used to supply heat for maintaining the cold room at a constant temperature.

Because of the complex nature of the operation of the test equipment and

the need for numerous readings as well as frequent adjustments to follow the prescribed temperature cycle, it was found desirable to make most of the control circuits automatic. The temperature balance between the inner and outer faces of the guarded box (D) is maintained with the balance circuit shown in Fig. 3. Single thermocouples with separate cold junctions are soldered to the center of each side of both the inside and the outside copper sheets. The four thermocouples on the outside are paralleled, as are the inner thermocouples and these groups are connected in series through the galvanometer (G). The galvanometer reflects a light beam on and off a photo electric

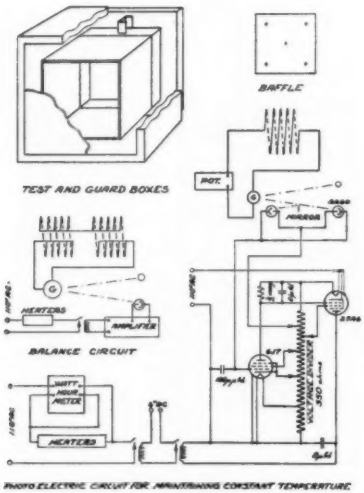

FIG. 3. AUTOMATIC CONTROL CIRCUITS

cell which operates a relay circuit to control the heaters for the guard surrounding the guarded box (D). When the guard side becomes warmer than the guarded box side the light beam is deflected onto the photocell and a portion of the heater units is turned off.

The necessity for an extremely sensitive thermostat for controlling the temperature in the guarded box became apparent early in the test. The sensitive bi-metallic thermostat used for steady state tests was not satisfactory because it had a tendency to block when the peak flow came through the wall. Five air thermocouples between the baffle shield (F) and standard panel (B) are used as the sensitive element of the thermostat circuit, shown in Fig. 3. These are connected in series with a potentiometer and a galvanometer. Deflection of the light beam by the galvanometer on and off the photocells operates the relays which turn a portion of the heaters on or off, as the case

may be. A two-stage vacuum tube amplifier is used to operate the relays.† The power consumed in the guarded box was metered with a specially built portable watthour meter. In order to prevent the light beam from overshooting the photocell and losing control of the temperature, a mirror picks up the light beam as it leaves the first cell and reflects it onto a second cell in parallel with the first. Stops on the galvanometers will also serve to prevent overshooting. When the voltage is properly adjusted to the heaters, this thermostatic device will control the air temperature to ±0.15 F indefinitely. With a potentiometer storage battery having a low discharge rate the poten-

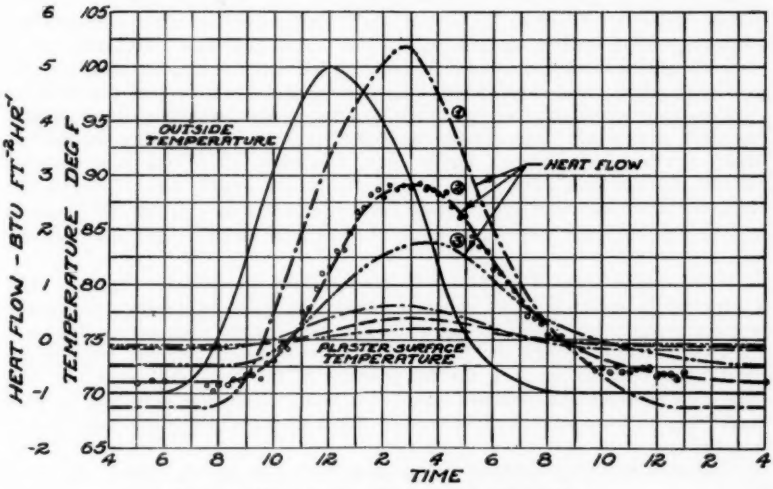
HEAT FLOW - BTU FT⁻²HR⁻¹
TEMPERATURE DEG F
TIME
OUTSIDE TEMPERATURE
HEAT FLOW
PLASTER SURFACE TEMPERATURE
6 5 4 3 2 1 0 -1 -2
105 100 95 90 85 80 75 70 65
4 6 8 10 12 2 4 6 8 10 12 2 4

FIG. 4. HEAT CYCLE RESULTS ON TEST WALL

1. Uninsulated; $C = 0.29$
2. Insulated wall—1 sheet of reflective insulation on paper, next to plaster base; $C = 0.17$
3. Insulated wall—3½ in. mineral wool; $C = 0.10$

tiometer needs to be standardized only twice in 24 hours. An additional advantage is that with the cold junctions in an ice bath and from a previous calibration of the thermocouples any desired temperature can be established in the guarded box (D) merely by setting the proper microvoltage on the potentiometer. Good operation of the device is dependent on low heat capacity heating units.

TEST PROCEDURE

The temperature cycle to which the outside of the test wall is to be subjected is either assumed or else determined by measuremnt of surface temperatures

† The amplifier used is described in "Application of Conventional Vacuum Tubes in Unconventional Circuits", by F. H. Shepard, Jr. (Proceedings of the *Institute of Radio Engineers*, Vol. 24, No. 12, December 1936, p. 1574.)

of existing walls under the desired conditions. The temperature of the air on the inner side of the wall is similarly fixed by the assumed air temperature that is desired. Fig. 4 shows a temperature cycle curve starting at 70 F and rising to a maximum of 100 F; the room temperature in the meantime remaining constant at 75 F.

The test routine is usually divided into four steps, (1) calibration of the standard panel to serve as a heat-flow meter, (2) measurement of conductance of the wall with a 25 to 30 deg temperature drop, (3) establishment of steady state heat flow through the wall for the initial temperatures with a conductance measurement for the small temperature difference prevailing under these conditions; this value should check the one obtained under (2); and finally (4) starting from the initial steady state condition the outside temperature of the wall is made to follow the precribed temperature cycle for a 24-hour period while the air temperature on the inner side is kept constant.

The standard panel is calibrated by operating the cold room and the guarded box at the required temperatures, which must be maintained constant during the test, and making the outside wall surface temperature the same as the inside surface temperature. Under these conditions all of the heat supplied to the guarded box (D), when balanced with its guard, flows through the standard panel, or meter (B). Since the temperatures in (D) and (C) remain constant through the subsequent tests the heat flow through (B) also remains constant.

If the conductance of the test wall has not been determined previously it is customary to measure it, as indicated under step (2), by raising the outside surface temperature to give a 25 to 30 deg temperature drop across the wall. Heat now flows through the test wall into the guarded box and reduces the heat input required to keep the temperature in the box, and hence also the heat flow through the standard panel, constant. From the reduction in heat input and the temperature drop across the wall its conductance can be calculated.

The next step is to establish steady state heat flow for the initial conditions of the test, in this case 70 F for the outside of the wall and 75 F air temperature in the guarded box. The conductance of the wall is again calculated from these data as a check though, because of the small temperature difference, the results are not as dependable as those obtained with the larger temperature difference, step (2). As soon as equilibrium is established the test is continued (4) by imposing the required temperature cycle on the outside of the wall. The temperature cycle is produced by varying the voltage on the exposed lights in chamber (A) with an auto-transformer. A cooling unit, which is operated continuously throughout the test, in the chamber (A) helps to produce the descending portion of the cycle by rapidly removing heat. The rate of heat removal is in excess of that required and the desired cycle temperature is maintained with the heater units. By this method a close and responsive control is maintained over the wall surface temperature. The cycle temperature is adjusted regularly every minute and all the temperatures as well as the power input to the guarded box (D) are read regularly in the same sequence at 15-min intervals. The principal purpose of the test is to determine accurately the heat-flow cycle through the plaster surface into the guarded box (D). The heat transmitted from the plaster into the guarded box is equal to the difference between the power input under steady state with

both wall surfaces at the same temperature and the power input at the time desired, times a conversion factor.

TEST RESULTS

Prior to making each cycle run the apparatus was calibrated with the wall in place and the plaster and the sheathing temperature at 75 F. Under the equilibrium condition no heat flows through the wall and all the input to the guarded box (D) flows through the standard blank. The sheathing temperature is then dropped to 70 F and held until the wall conductance is rechecked. An arbitrarily chosen temperature cycle, Fig. 4, was applied to the test wall. The conventional wall was tested under three different heat resistance conditions. First, the plain wall without insulation; second, with a medium amount of reflective insulation of negligible density in the air space; and third with a large amount of relatively low density blanket insulation in the air space.

A sample calculation for the heat flowing from the wall is given as follows:

For the heat flow at 4:00 P.M. for the reflective insulated wall.

4:30 P.M. power	728.18
3:30 P.M. power	646.75
	81.43

Steady state power 91.98 (heat flow through standard panel).
Heat flow = (91.98 − 81.43) × 0.2561 = 2.70 Btu per square foot per hour.
This value was plotted at 4:00 P.M.

The value from 3:45 to 4:45 is plotted at 4:15 p.m. The values plotted thus give the same curve as the 15-min power differences and they show less scattering. The heat-flow curves and plaster-surface temperature curves plotted in Fig. 4 show the magnitude and time displacement of these factors.

During the test the sheathing and plaster-base-surface temperatures facing the studs were also read at 15-min intervals. These curves are omitted from the plot to avoid confusion. A proportionate number of thermocouples placed over and under the studs give due weight to the effect of heat flowing through these members on the various surface temperatures. A check of the results may be obtained by the plaster-surface coefficients while heat flows out of the plaster surface. When the average heat flowing into the guarded box (D) from the wall is divided by the average plaster-to-air temperature difference above 75 F the average still air-surface coefficient is obtained. The values for the walls tested are shown in Table 1.

These values are in reasonable agreement with the value 1.34, that is usually obtained in the guarded hot box under still air conditions. They also serve to confirm the validity of the results of a particular test.

The reliability of the conductance values measured by the modified guarded hot-box method is seen in the alignment of points, with respect to the mean temperatures, for the blanket insulated wall. Two tests were made on the wall at different mean temperatures by the conventional guarded hot-box method. The third test at the highest mean temperature was made by the

modified guarded box method on the wall insulated in the same way as for the first two tests but with a different sample of insulation. The values are as follows:

Mean Temperature, F	Conductance
35.9	0.077
50.5	0.087
88.2	0.111

A plot of these values, conductance *vs.* mean temperature, shows excellent alignment of points and confirms conductance tests made with the apparatus assembly shown in Fig. 1.

The question may arise as to the accuracy of results where the heat flow depends upon the small differences between large numbers. In several cases cyclic tests have been repeated on test panels to determine the reproducibility of the results. In all cases the heat flow curves were the same. This may be construed as evidence that the automatic controls will function properly. It is felt that the heat flows measured by the modified guarded hot box are accurate because of the agreement between the conductance values obtained by the modified and regular guarded hot-box methods. The fact that the surface coefficients are also in close agreement with the steady state values is evidence that the method of measuring the heat flows in the modified guarded hot box is reasonably accurate.

APPLICATIONS AND LIMITATIONS OF THE APPARATUS

In a wall or roof where heat flows under steady state conditions through all the component parts in series, it is a simple matter to calculate the resistance from our present knowledge of the coefficients of the individual parts. When there are parallel paths of heat flow in the structure it is still relatively simple to calculate the resistance of the wall under steady state conditions. Although the calculations are based exclusively on the steady state, such a condition seldom prevails for any extended period of time. The wall or roof is usually subjected to some sort of a temperature cycle which may make the steady state calculations deviate widely from the actual condition, particularly where the cycle is daily as in the case of air conditioning. As previously mentioned this condition adds to the complexity of the problem and makes calculations difficult. On account of the limited knowledge we have on heat diffusivity of materials in series and parallel combinations it is almost impossible to calculate

TABLE 1—VALUES FOR WALLS TESTED

WALL	CONDUCTANCE	AIR TEMPERATURE, F	AVERAGE PLASTER TEMPERATURE, F	f_i
No Insulation	0.288	74.7	75.7	1.48
Reflective	0.174	75.0	75.6	1.44
3⅝ in. Blanket	0.101	74.8	75.1	1.49

heat flows in a wall under transient states. The test method for such conditions as described in this paper is relatively simple and should add much to our present knowledge of heat transmission.

If field observations are made on outside wall and roof temperatures, the cycles can be reproduced in the laboratory on similar constructions. In this way the effect on the heat-flow cycle of such factors as sudden cloudiness, scattered clouds, showers and shading of walls and roofs by other buildings can be studied. The method is also applicable for studying the relation between the physical and thermal properties of insulations on the heat transfer for transient or cyclic temperatures.

The cyclic temperature effects during the heating season are not as important as during the cooling season. There are, however, conditions that arise that might be investigated. During the winter sudden temperature changes due to storms or sunshine on walls and roofs, impose severe temperature cycles badly disrupting the assumed steady heat flow. These could also be investigated by providing sufficient cooling in chamber (A) facing the outside of the wall and reproducing the cycle with the unshielded lamps.

It should be noted that in cases where the resistance of the wall to heat flow is very low and the peak of the imposed temperature cycle is high the heat resistance of the standard blank must be reduced accordingly. There must always be as much or more heat input to the guarded box (D) under steady state as flows through the panel with the peak flow, otherwise the temperature within the guarded box (D) will build up with a summer cycle.

It is also possible to test roof sections in the vertical position if they do not contain air spaces. The temperature gradient on the guarded box side of the test section is made very small by locating the heaters at the bottom of the boxes. On the cycle side the surface temperature is made the same all over with the fans. If there are air spaces within the roof, the test panel and apparatus would have to be inclined to a position corresponding to that for which the field temperatures were taken.

If sunshine is a factor which determines the temperature cycle to be imposed on the outside of the wall, this method cannot be used unless the wall is opaque. The short wave length solar radiation which passes through such materials as glass cannot be produced in the laboratory. If windows are shaded by an awning from direct sun radiation they may be exposed to considerable reradiated and solar energy from surrounding objects. This condition can be investigated in the apparatus described.

ACKNOWLEDGMENTS

The authors wish to acknowledge the assistance given them by the U. S. Navy, for whom the apparatus and test method herein described were developed. The assistance given by E. R. McLaughlin, research assistant, Engineering Experiment Station, The Pennsylvania State College, in making these tests is also acknowledged.

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DISCUSSION

W. S. STEELE[10] (WRITTEN): I believe that the apparatus is well designed and that it will accomplish the purpose for which it is intended.

From a commercial standpoint, however, I view the general adoption of this test apparatus as dangerous, because a report of an individual test cannot be construed as representative of the results which might be obtained by the home owner over any extended period of time. It is designed to test insulating walls, and yet it does not consider the possibility of condensation and its reaction upon the insulation.

I believe that the data to be gained from tests utilizing this equipment will be valuable for the manufacture and for the research department, but I believe that the use of a single test containing a specific heat cycle will be dangerous if used by lay salesmen.

P. F. MCDERMOTT[11] (WRITTEN): New test methods for measuring heat transfer are rare and are of great interest. The method described is not only new but very ingenious and, therefore, that much more interesting.

New methods should be carefully studied and analyzed and should be subjected to criticism from every possible angle until they have proven themselves to be reliable. The method of the authors seems to defy criticism provided its operation is in a well equipped laboratory and its direction by a competent operator. In other words, the major criticism appears to be of the complexity of the apparatus and the great care necessary to carry through the several steps of the procedure without introducing error.

Information on heat transfer resulting from non-steady temperature gradients is badly needed to aid in the design of various structures since many of them, particularly buildings, are generally in non-steady conditions. It is believed that more

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[11] Research Engineer, Johns-Manville Research Laboratories.

and more of these data may be obtained in the future and now seems an opportune time to suggest that the type of cycle be standardized. The selection would first be between one to simulate the diurnal variation of some average location and exposure and one to permit simple mathematical analysis. The cycle used by the authors is an example of the former while a simple sine curve might be the best example of latter. In either case both the amplitude and period would need be considered. All these factors bear on the results obtained from the tests. We suggest that this matter be taken up by the Technical Advisory Committee on Insulation with the aim of making recommendations for a test cycle.

With respect to the data reported in this paper, there are several items that we would like to have described in greater detail. It has been observed that the temperature of cold junctions frequently vary as much or more than the ± 0.15 F stated to be the limits of temperature control by the thermo-electric regulating device. Information as to how this variation was avoided by the authors should be of interest to those interested in tests and methods.

A very interesting phase of cyclic heat transfer is that associated with the influence of previous cycles on the particular one under observation. For example, in studying the absorption of solar energy it was found that unless several consecutive clear days with their resulting similar influences were had prior to the day being studied, an analysis of the data could not be made satisfactorily. This is due to the phenomena of reflected waves within the structure acting as harmonics of the initial wave applied to the structure. Whether the data reported by the authors on each wall were on a first cycle or on one which followed several similar previous cycles would be of considerable interest. If several successive cycles were investigated on each wall, data on all of them would be of value.

F. B. ROWLEY (WRITTEN): The authors have presented a unique adaptation of the hot-box test method for conditions of variable heat flow through a wall. As ordinarily applied the hot box is adapted to either constant or variable heat flow so long as the heat flow through the wall is in one direction. The present method solves the problem of measuring the heat flow throughout a cycle during which the direction of flow through the wall may be reversed.

I agree with the authors that there is a specific need for evaluating different types of construction under variable heat flow conditions. This is particularly true for summer cooling problems. A point of particular interest is the heat-lag curves which are shown in Fig. 4. It would be desirable for the authors to extend their investigation to include all of the typical walls used in building construction. Heat lag is an important factor in designing summer cooling equipment and more fundamental test data would be assuring for the estimating engineer.

A point which is not quite clear to me is the apparent disagreement between the test values which they have obtained by this method on the three walls and the values for corresponding construction as given in THE GUIDE. It is true that these were obtained at different mean temperatures than those values given in THE GUIDE, but relatively it seems to me they should be the same.

If we take the values given under Fig. 4 for Walls 1, 2, and 3 the following is obtained:

WALL NO.	CONDUCTANCE, *C*	RESISTANCE, *R*
1	0.29	3.45
2	0.17	5.88
3	0.10	10.00

As I understand it, these walls are all of the same construction with the exception of treatment of space between the studs. All of the walls are constructed with rock lath as a plaster base. In Wall 1 the inner surface of the rock lath is lined with paper; in Wall 2 it is lined with metal foil; and for Wall 3 the surface is lined with paper but the space between the studs is filled with 3⅝ in. mineral wool. If this is the case, the difference in the conductance values should be proportional to the difference in the resistances from surface to surface of air space for the various walls. If we take values for the three types of construction and correct the resistance values for the area occupied by the studs we have the following results as the resistance values from face to face of studs.

Wall 1—Air space at 80 F.

$$a \text{ (from THE GUIDE)} = 1.27 \quad \text{Resistance} = 0.787 \times \frac{14.375}{16} = 0.707$$

Wall 2—Air space warm side lined with metal foil at 80 F.

$$a \text{ (estimated)} = 0.50 \quad \text{Resistance} = 2.00 \times \frac{14.375}{16} = 1.798$$

Wall 3—Space between studs filled with 3⅝ in. of mineral wool at 80 F. mean temperature.

$$k \text{ for wool} = 0.29 \text{ (from THE GUIDE)} \quad \text{Resistance} = 12.5 \times 14.375 = 11.23$$

If Wall 1 is used as a basis the calculated resistance, R, for Wall 2 is,

$$3.45 - 0.707 + 1.78 = 4.541, \text{ and} \quad \text{Conductance, } C = 0.220$$

For Wall 3 the resistance, $R = 3.45 - 0.707 + 11.23 = 13.97$, and

$$\text{Conductance, } C = 0.072$$

These values are so far different from those given under Fig. 4 as to suggest the desirability of a more complete investigation to determine the cause of the variation.

W. T. MILLER (WRITTEN): The practical value of laboratory determinations for cyclic flow is somewhat vague as there are a number of uncontrolled variables encountered under practical conditions all of which effect heat flow.

Two of the nine experimental houses on the Purdue Housing Research Campus have been equipped with the necessary installations to determine a complete temperature gradient through each of the four outside walls. Experience with these walls emphasizes some of the difficulties encountered by the heat during its relentless journey.

Sun and wind do peculiar things to the apparent surface conductance. In the laboratory, there is a reasonably close relationship between surface conductance and air velocity, while on a building there seems to be no correlation. The wall exposed to the wind may or may not have a greater apparent surface conductance than the lee wall. A specific illustration is taken from data from a North wall and an East wall of the same room. The data were obtained in the afternoon of a sunless day while the wind blew from the Northwest at 10 mph. The apparent surface conductance of the North wall was 4.96 and for the lee wall, East wall, it was 7.36. Again, from the same walls, an apparent surface conductance of 1.03 was evident on the North wall and the surface conductance for the East wall was apparently 0.82. The data were obtained during the forenoon of a sunless day with an East wind of 7 mph. While the sun was not visible, it nevertheless had an effect on the East wall surface.

Gradients of the West and East walls were obtained at sunset, and are here presented. The room temperatures were observed at the breathing line at the center of the room while the gradient air temperatures were obtained at points 6 in. from the surface.

TIME	WALL	OUTDOOR AIR		OUTSIDE SURFACE	INSIDE SURFACE	INSIDE AIR	
		Amb.	Grad.			Grad.	Room.
5 P.M.	W	65	65	98	74	72	72
5 P.M.	E	65	65	68	70	71	72
6 P.M.	W	58	58	78	75	72	72
6 P.M.	E	58	58	65	67	68	72
7 P.M.	W	48	48	49	70	70	72
7 P.M.	E	48	48	49	64	67	72

Thus it may be seen that it would indeed be difficult to obtain stable field data for laboratory duplication.

COL. W. A. DANIELSON (WRITTEN): It can be safely said that it is beyond the power of the normal human mind to apply Fourier's Series to heat flow. There are so many variables that any result is doubtful until it has been proven by actual test. I am sure that this study by the authors is much appreciated by the heating engineers.

No doubt, all of us have experienced the condition where a building is hotter at midnight than it is during the day. This is in spite of the fact that the outside air is cooler. The test data set forth in the article gives the reason. In this particular article, the lag between the highest outside temperature and the greatest heat flow to the interior is 3 hours, and this is true even with a relatively well insulated wall. In the case of a wall that has more mass and, therefore, greater heat storage capacity, such as solid brick, the lag would undoubtedly be much greater and the total heat much greater. It is hoped that test will include such walls.

It is interesting to note that the slight temperaure differences on the plaster surfaces produce marked differences in heat flow. Undoubtedly this is due to convection currents on the surface of the plaster. It would be very interesting to make some tests with the convection current broken up by a honeycomb with openings normal to the surface in test. A few experiments will determine the openings and depths of the honeycombs, and the distance that these should be from the surface in test. Points of measuring temperatures must also be determined in a similar way.

Here in the tropics reradiation mentioned in the next to the last paragraph of the paper is very marked. The afternoon heat emitted from roads and other exposed surfaces with a high heat storage results in the most uncomfortable part of the day. The afternoon siesta, which is so usual with the native inhabitant, is undoubtedly the result of this reradiation. Sleeping rooms are closed off so that no outside radiation strikes the sleeper but air is permitted to circulate through.

M. M. MCCLURE:[13] I should like to give some of the experiences that we have had in our hot-box work at Purdue University. We have two hot boxes, one we call our small hot box, which is 5 ft square and has a 3 sq ft test area. In this box we can test either in a horizontal or a vertical position, and it is also equipped so we can accurately control the air velocity over the cold side.

We have run tests with still air in which the velocity is so low that we can scarcely detect it by use of a velometer. Other tests have been made with controlled velocities up to 15 mph. We have been very successful in doing this. We have also tested samples in horizontal position. In this position we can also control the air flow over the cold side of the test specimen.

Our other guarded hot box, usually referred to as the large box, requires a test sample 7 ft x 11 ft. The test area of this box is 4 ft x 8 ft, being large enough

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to test full sized wall panels of the prefabricated type. The box has been used in an extensive series of tests of the treatment of railway refrigerator cars. Another comprehensive series is being continued in the study of wall space treatment in which a wide variety of types and thicknesses of insulation have been applied to a 3⅝ in. This box is full automatically controlled, a continuous heat supplied being furnished to the hot box while the guard-box heat is supplied intermittently, the guard-box temperature being a function of the hot-box temperature.

There is one thing we have never done at the University in our hot-box work, and that is, used fans to eliminate stratification in our boxes. Our boxes, heaters and baffles are designed in such a manner so as to break up the stratification without the use of fans. In 1934, when I started on the hot-box work, we used fans one summer and then tried to duplicate our work, and we were never able to do so. After that we discontinued fans and have spent a great deal of time on installation of heaters so that we would be able to break up stratification without the use of fans. We are very happy in the outcome, because we are able to duplicate results readily, and feel that the fact that we have not used fans removes one important variable.

There is another point that we have found in the proposition of surface conductance. We ran a series of tests in which we found a discrepancy between the surface conductance on the warm side and surface conductance on the cold side. We could measure our air velocities over the cold side, but we could not, because of structural difficulties, measure the air conductances on the warm side. Our warm side conductances were a little higher than those of our cold side, so to satisfy ourselves, we were going on the assumption that we probably had a little more velocity over the warm side than we had over the cold side.

Last year we had a graduate student who was doing some research work in surface conductances, and in his tests he was checking air velocities over a warm surface. Its surface was warmer than the air. The heat flow was from the surface to the air, and he found that in that case he checked our seemingly low surface conductances. Then he reversed his process and used a warm air flowing over a cooler surface, and found that in that case his surface coefficient ran a little higher, which substantiated the findings we had made two years before that.

It made us feel rather good, because we found out then the information we had, which seemed somewhat questionable, was given this substantiation by this work of one of our research fellows.

Professor Queer has told you how he has controlled the surface temperatures to a very close degree of accuracy, by the use of thermocouples. He is more fortunate than we have been. We set up our boxes with surface thermocouples as controls, but were unable to do the job successfully. But with the use of resistance thermometers made of nickel wire, which are suspended an inch from the surface, both inside and outside of our guarded hot box, used as legs of Wheatstone Bridge, we have used them as an anticipator, and have been able to control our wall temperatures to three or four thousands of a millivolt. With this set-up we are able to maintain these without any attention over periods as long as we care to. In some cases we have maintained this for periods as long as a week without any attention.

No. 1179

REHEATING BY MEANS OF REFRIGERANT COMPRESSOR DISCHARGE GAS

By S. F. Nicoll* (*MEMBER*), York, Pa.

SUMMER air conditioning usually requires the simultaneous removal of sensible and latent heat from the treated air. The sensible cooling and moisture removal are sometimes accomplished independently, but more often a spray dehumidifier or a refrigerated coil performs both functions at once. This paper deals with this combined sensible and latent cooling and with the means for varying the relative proportions of each, particularly with regard to the usefulness of compressor discharge gas for this purpose. The discussion will be limited to coils as the cooling means, although the principles set forth apply also to spray dehumidifiers.

It is believed that the principles surrounding the use of condenser discharge gas reheat and the discussion of reheat generally as set forth in this paper will be of interest to the air conditioning industry.

Basic Principles

It is not within the scope of this paper to discuss methods of rating coils. A review of some of the fundamentals, however, provides an understanding of the phenomenon of simultaneous sensible and latent cooling. If a straight line is drawn on a non-logarithmic psychrometric chart through points representing the conditions of the air entering and air leaving the cooling coil, its slope will be a measure of the so-called *load-ratio* and the line can be referred to as the *load ratio line*. The load ratio may be expressed in various ways but, for the purposes of this paper, will be considered as the ratio of total to sensible heat removal. If the load ratio line is extended to intersect the saturation curve, there will result a relationship known as the *straight line law*. This law states that the point of intersection with the saturation curve can be considered the *coil surface temperature*. A straight line drawn through points representing the entering air condition and the coil surface temperature will then be the load ratio line and the leaving air condition must fall also on this line. A reference to Fig. 1 will show that, for a given entering air condition, the load ratio and therefore the proportion of latent cooling, will

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Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, Kansas City, Mo., January, 1941.

increase as the coil surface temperature decreases. Although somewhat of an approximation, it is generally assumed that the load ratio can be increased by decreasing the coil surface temperature only until the load ratio line becomes tangent to the saturation curve as, for instance, line *A-B* in Fig. 1. If a greater load ratio is wanted, such as represented by line *A-C*, it will be necessary to have recourse to reheat to neutralize the unwanted sensible cooling which will accompany the desired dehumidification. For instance, air may be cooled so that the leaving temperature falls at point *D* on the 1.58 load ratio line and then reheated from 55 deg to 75 deg so as to fall at point *E* on the desired load ratio line *A-C*. Again, air may be treated so that its leaving temperature falls at point *F* on load ratio *A-B*, and then reheated to point *G*. The air quantity and the choice of conditioning equipment will determine whether path *A-D-E*, or *A-F-G*, or some other is chosen.

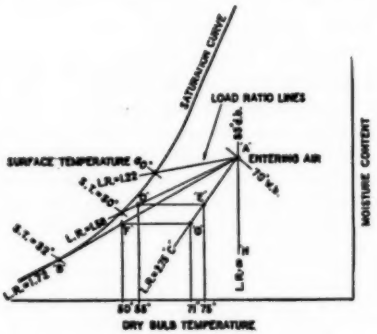

FIG. 1. CHART FOR DETERMINING LOAD RATIO

If sufficient reheat is applied to neutralize all sensible cooling, there will be dehumidification only. The load ratio, as defined, is then infinite and the load ratio line will be *A-H*.

The selection of cooling equipment is based on chosen definite maximum sensible and latent loads. Because of variations in weather conditions, internally generated heat, etc. the apparatus is seldom presented with a required load or load ratio equal to that upon which the original selection was based. Cases often arise necessitating a greater load ratio at a light load than at the maximum total load, because of a disappearance of part of the sensible load without a corresponding decrease in the latent cooling requirements. Unless some provision is made to accommodate the capacity and load ratio of the cooling equipment to the requirements, the dry-bulb temperature will under these circumstances fall to the minimum permissible and the apparatus will be stopped by the controlling thermostat. This results in the undesirable condition of high humidity at light sensible loads so common with the conventional comfort conditioning system without bypass, reheat or other means for satisfactorily altering the load ratio with changes of load. Under these circumstances also, the cooling equipment operates intermittently and tends to

cycle, on and off, at short intervals, which has a very adverse effect on the relative humidity. During the *off* periods untreated outside air, usually of relatively high moisture content, is admitted. Any moisture on the coils or undrained from the apparatus tends to re-evaporate into the air stream. A considerable portion of the operation may take place at a high coil surface temperature and therefore low load ratio, further aggravating the humidity condition. To improve the humidity situation some arrangement should be used

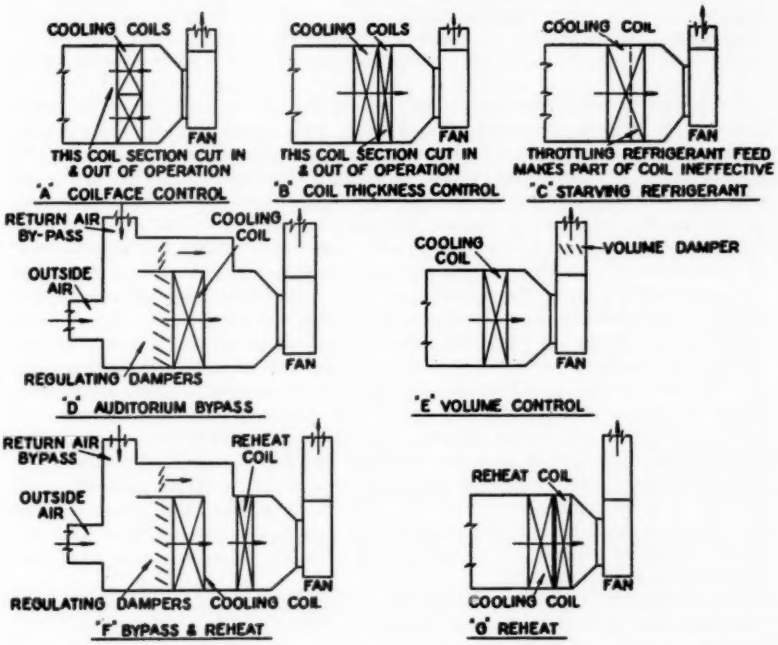

FIG. 2. TYPICAL ARRANGEMENT FOR REGULATION TOTAL CAPACITY IN LOAD RATIO

to modify the equipment load ratio as required and to avoid short cycling operation. This is, of course, an absolute necessity on industrial installations where definite temperatures and humidities must be maintained.

REGULATION

A number of arrangements are in common use to accommodate the apparatus performance to changes in load. The degree of effectiveness varies. Some alter the equipment load ratio accurately, while others are advantageous chiefly in keeping refrigeration on the coils, or part of them, at light loads and avoiding therefore the adverse effects of rapid *on* and *off* control. The most usual types of control are, unloading the compressor, coil face control,

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TABLE 1—TYPICAL REQUIREMENTS FOR COMFORT COOLING INSTALLATION

	MAXIMUM LOAD	MINIMUM LOAD
Air Entering Coils....	83.5 deg dry-bulb and 69.1 deg wet-bulb	74.2 deg dry-bulb and 64 deg wet-bulb
Inside Conditions....	80 deg dry-bulb and 50 per cent relative humidity	74 deg dry-bulb and 50 per cent relative humidity
Total Sensible Load..	94,800 Btu per hour	40,300 Btu per hour
Total Latent Load...	36,000 Btu per hour	39,200 Btu per hour
Total Load.....	130,800 Btu per hour	79,500 Btu per hour
Load Ratio.....	1.38	1.97

coil thickness control, starving the refrigerant feed (with volatile refrigerant), air volume control, the auditorium bypass, and reheat (either alone or in combination with certain of the others). Compressor unloading may be accomplished by change of speed, stopping individuals of a group or by capacity reducing devices. With face control the cooling coil is divided into several parts, parallel as to the air flow, one or more of which can be put in and out of operation by the control (see Fig. 2*A*). When the entering air is a mixture of recirculated and outside air, this arrangement admits some untreated outside air to the conditioned space when the controlled coil section is off. Coil thickness control consists in varying the number of rows deep in operation, as shown in Fig. 2*B*. This is equivalent to the starving of the refrigerant feed as shown in Fig. 2*C*, when the refrigerant feeds through the coil parallel to the direction of air flow, as is common. The auditorium bypass is shown in Fig. 2*D*. As far as the cooling equipment is concerned, both the bypass and volume control, Fig. 2*E*, act to reduce the air quantity through the cooling coil and for purposes of this discussion are therefore considered equivalent. Reheat is commonly used in two arrangements, Figs. 2*F* and 2*G*. A modification of Fig. 2*F* consists in placing the reheat coil in the bypass instead of between cooling coil and fan as shown (see Fig. 4). Other means of control are sometimes used but the foregoing will serve the purposes of this discussion.

The comparative performance of the arrangements shown in Fig. 2, and especially the part that can be played by reheat, is best illustrated by a specific example. Consider a comfort cooling installation for which the cooling equipment selection is made for the maximum load given in Table 1. Suppose that it is desired to hold inside conditions of 74 deg and 50 per cent when the minimum load indicated in Table 1 prevails. The values in Table 1 are based on arbitrary assumptions of outside air quantity and condition as well as population and internally generated heat, but are sufficiently typical for illustrative purposes.

A 10 hp Freon-12 water cooled condensing unit and a six row cooling coil, of typical design, handling 4500 cfm, will satisfy the maximum load requirements given in Table 1. Table 2 shows how the equipment as originally selected and variously modified will perform under the minimum load conditions.

Table 2 shows that all of the arrangements listed will satisfy the sensible load, but their effectiveness in removing the necessary moisture varies widely. Those with a load ratio less than the required 1.97 will not succeed in producing the required relative humidity because the running time will be fixed by

the thermostat to suit the sensible load and the latent heat abstraction will be insufficient. *Cases* 1 to 5 cannot therefore attain the desired conditions. A line on the psychrometric chart tangent to the saturation curve and through the minimum load entering air conditions will have a slope equivalent to a load ratio of 1.70. As described previously, then, it will be considered impossible, with air entering at 74.2 deg dry-bulb and 64 deg wet-bulb, to attain a load ratio in excess of 1.70 without reheat. It is of interest to observe how closely this limit can be approached by the various non-reheat arrangements shown in Fig. 2. In *Case* 2, the rebalance between the original coil and the reduced compressor displacement will result in so high a coil temperature as to greatly reduce the proportional dehumidification. *Cases* 3 and 4 are much better in this respect, whereas the auditorium bypass, *Case* 5, is still more effective. Quite aside from the performance shown in Table 2, *Cases* 2 to 5 have the advantage over *Case* 1 of longer running time before the low limit of temperature is attained and, therefore, minimize the disadvantages of *on* and *off* operation described previously. This situation is not susceptible of even reasonable mathematical treatment and no modification for it appears in Table 2. Although frequently used on comfort installations no non-reheat case would be effective on an industrial installation, where a definite temperature and humidity must be maintained.

It is true, of course, that Table 2 would change somewhat if the requirements, equipment selection, and modifications were varied. It is believed, however, that the situation as presented is sufficiently representative of practical conditions to justify the conclusions. At any rate, without the use of reheat it is impossible to attain the required load ratio of 1.97.

A further reference to Table 2 will show that in *Cases* 1, 4, and 5, there is sufficient sensible and latent capacity but there is a proportional excess of the former. If reheat be applied to the air stream leaving the cooling coil, there will result *Cases* 6, 7 and 8, in which the required conditions will be

TABLE 2—REHEATING EQUIPMENT PERFORMANCE AT LIGHT LOAD CONDITION

	Required	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
Total load Btu per hour	79500	118300	73200	77900	106300	100800	79600	79100	81400
Sensible load Btu per hour	40300	79100	61000	51900	67300	60700	40400	40100	41300
Latent load Btu per hour	39200	39200	12200	26000	39000	40100	39200	39000	40100
Load ratio	1.97	1.50	1.20	1.50	1.57	1.66	1.97	1.97	1.97
Reheat Btu per hour		None	None	None	None	None	38700	27200	19400
					Total Compressor Load		118300	106300	100800

NOTES: *Case* 1—Original equipment without change rebalanced against minimum load conditions.
Case 2—Compressor speed reduced to half.
Case 3—Coil face reduced to half, Fig. 2*A*.
Case 4—Coil reduced from 6 to 4 rows deep, Fig. 2*B*. This is equivalent also to starving the coil of refrigerant, Fig. 2*C*.
Case 5—Auditorium bypass handling 2400 cfm recirculated air, Fig. 2*D*. This is equivalent also to volume control.
Case 6—*Case* 1 with reheat as indicated.
Case 7—*Case* 4 with reheat as indicated.
Case 8—*Case* 5 with reheat as indicated.

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attained. Table 2 shows also that *Case* 7, and to a still greater extent, the auditorium bypass in *Case* 8, are effective in reducing the refrigeration and reheat requirements as compared to *Case* 6.

SOURCES OF REHEAT

The reheating may be accomplished by various means. Office buildings, stores, and hotels frequently have steam available in summer and in the spring and fall seasons, which they must waste at low pressure and which is conveniently available for reheating purposes. There are even comfort cooling installations where, to keep the humidity down, the winter heating system has been placed in operation as a means of reheat. Steam, hot water, electricity or refrigerant compressor discharge gas may be used for reheating.

When a compressor and coil are operating to cool and dehumidify air the heat extracted from the air, together with the heat equivalent of the horsepower expended upon the refrigerant, must be rejected in the condenser. Moreover, the cooling effect is the sum of sensible and latent heat abstraction. If the compressor discharge gas is used for reheating purposes there will be available, in all cases, much more than enough rejected heat to counteract the entire sensible cooling effect of the equipment. Although there is rejected heat available to raise the dry-bulb temperature of the air above that at which it enters the conditioning equipment, it has not been common, in installations so far made, to do more than to simply neutralize the whole sensible cooling effect. It is to be noted that, since the source of reheat is the refrigerating system itself, there will be no heat available for reheating unless the cooling equipment is already running for dehumidifying purposes.

By the use of compressor discharge gas there is automatically made available a source of reheat without incurring the expense of steam, hot water, or electricity for that purpose. On the majority of air conditioning installations, steam or hot water are either not available at all during the cooling and dehumidifying season, or can be produced only by operation of the winter heating plant specifically for reheating purposes, often an inconvenient and expensive situation.

Compressor discharge gas has another advantage as against other sources of reheat. Because of the effectiveness of the reheat coil as a refrigerant condenser, the condensing water requirements are cut down, thus reducing operating expense.

COMPRESSOR DISCHARGE GAS REHEAT

Fig. 3 illustrates a typical arrangement for compressor discharge gas reheat. The reheater is simply a condenser coil located in the air leaving the cooling apparatus. This condensing, or reheat coil, is arranged in parallel with the main condenser, which may be water or air cooled or of the evaporative type. The reheat coil requires some form of control such as the valve shown, as well as a means for disposing of the condensed liquid; that is, some equivalent of the steam trap that would be used with a steam coil. It is apparent that reheat will generally be used at times when the total refrigerating load is less than the maximum. Besides, the reheat coil acts as additional condensing

capacity over that provided in the condensing unit. A fall in condensing pressure, and corresponding temperature will result when the reheat coil is in operation unless some provision is made to prevent it. This fall of condensing temperature will reduce the reheat capacity and may also result in too low a differential pressure between high and low sides of the refrigerating system to handle the required liquid through piping, thermal expansion valves, etc., and against any static head that may exist. It is frequently necessary, therefore, to provide some means for maintaining the condensing pressure at

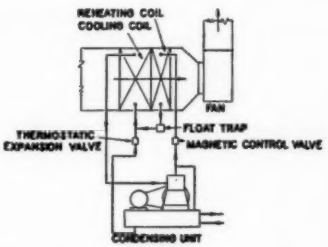

FIG. 3. ARRANGEMENT FOR COMPRESSOR DISCHARGE GAS REHEAT

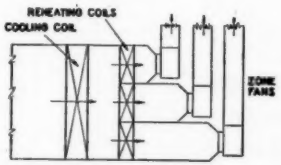

FIG. 5. REHEAT COIL ARRANGEMENT FOR INDIVIDUAL ZONES

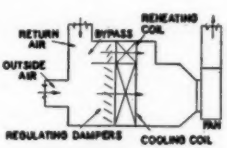

FIG. 4. ARRANGEMENT WITH REHEAT COIL IN BYPASS

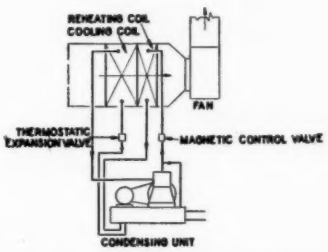

FIG. 6. THE EQUIPMENT ARRANGEMENT WHEN DISTANT REHEAT COIL IS USED

some design value. This value may be lower than that expected at peak load conditions. On a water cooled condenser, a pressure actuated water valve will serve this purpose. On evaporative condensers the water circulating pump may be operated intermittently, the air supply throttled, or both, by a control device sensitive to the condenser pressure.

The cooling coil and reheat coils bear certain relationships to each other. The liquid condensed in any reheat coil should be fed to the cooling coil in the same air circuit. If, for instance, there were several cooling systems with reheat coils on a common compressor and the principles previously described were not observed some reheat coil might feed more liquid into some cooling coil than it could evaporate, a condition which could not be tolerated. If properly arranged, on the other hand, the reheat coil will condense a liquid quantity only sufficient to neutralize the sensible cooling of the air and, since the cooling coil removes both sensible and latent heat from the same air, there

will always be less liquid furnished to the cooling coil by the reheat coil than the total cooling load requires. The thermostatic expansion valve, as shown in Fig. 3, will then control the liquid feed so as to maintain the required suction superheat.

Several arrangements of the air treating apparatus have been used. The reheat coil may be in the air leaving the cooling coil, as shown in Fig. 3, or in the bypass (see Fig. 4). Several reheat coils may serve individual zones

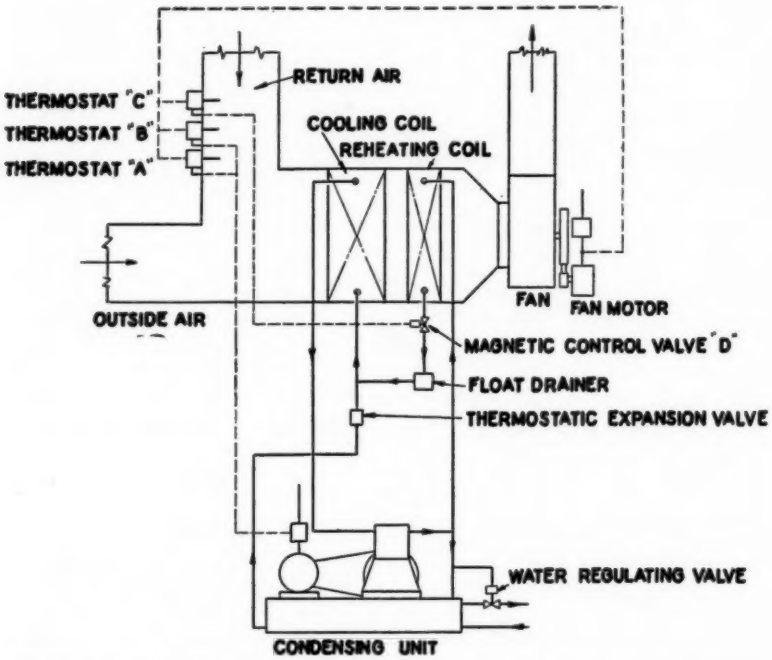

FIG. 7. LAYOUT OF SUMMER COOLING INSTALLATION WITH REFRIGERANT COMPRESSOR GAS REHEAT

as in Fig. 5. It is possible also to place the reheat coil as indicated in Fig. *2F*. A reheat coil when used in combination with a bypass, as in Fig. *2F*, and particularly as in Fig. 4, will be exposed to an appreciably higher air temperature than the reheater in Fig. 3. This is compensated for to some degree by the comparatively greater reheat that the Fig. 3 arrangement will require for a given effectiveness in changing the load ratio. Compare, for instance, *Case* 6 with *Case* 8, in Table 2.

Reference to Table 2 will show that the total compressor load, and therefore the operating expense is less with *Case 7* than with *Case* 6 and still less with

Case 8. The bypass, *Case* 8, is therefore an effective means for reducing operating cost on reheat installations of this type. It does, however, present added first cost and complication and is not conveniently arranged in many air conditioning units. For these reasons an analysis of the requirements has resulted in the choice of the arrangement illustrated by *Case* 6 and Fig. 3 in most instances.

The most usual refrigerant hookup is shown in Fig. 3. If, however, the

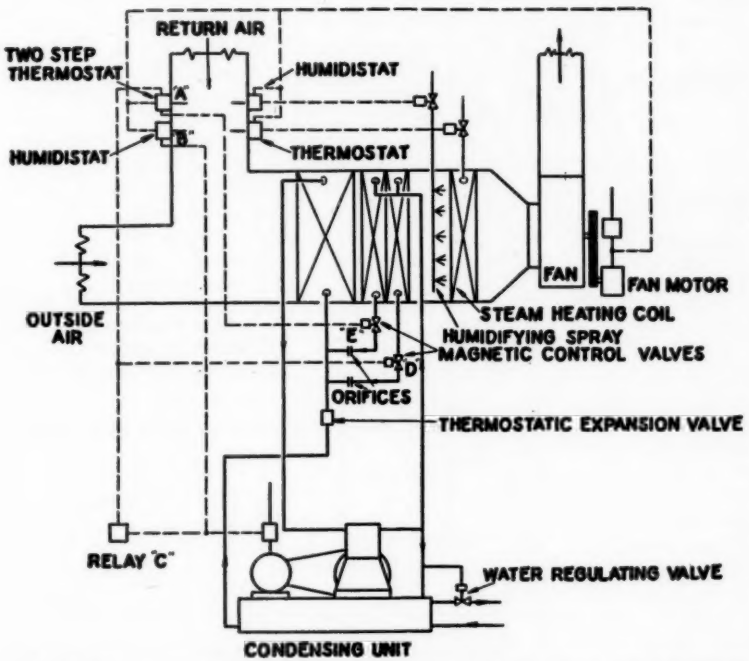

FIG. 8. YEAR 'ROUND INSTALLATION WITH DIVIDED REHEAT COIL FOR GRADUATE CONTROL

reheat coil is a considerable distance above the high pressure liquid receiver it is possible to use the arrangement shown in Fig. 6. The condensed liquid is returned to the high pressure receiver by gravity instead of to the cooling coil by pressure difference. The return pipe enters the receiver below the liquid level and the liquid stands in this pipe to a height corresponding to the pressure drop through the reheat coil and the connections to and from it.

Usually a reheat coil is selected with a capacity sufficient to counteract all sensible cooling, so that the equipment can serve to dehumidify only when required; thus it can be termed *complete reheating*. On the other hand, for comfort applications particularly, there have been instances of less than complete reheating because of apparatus or space restrictions or other reasons.

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This may be termed *partial reheat,* and serves to increase the apparatus load ratio beyond the limit described as line *"A-B"* in Fig. 1, but not up to the point of dehumidification alone without sensible cooling. A reheat coil of this type in an auditorium bypass simply increases the effectiveness of the bypass as a load ratio corrector.

Automatic control may be accomplished in various ways. If a reheat coil is located in an air bypass, the capacity is varied by the action of the bypass

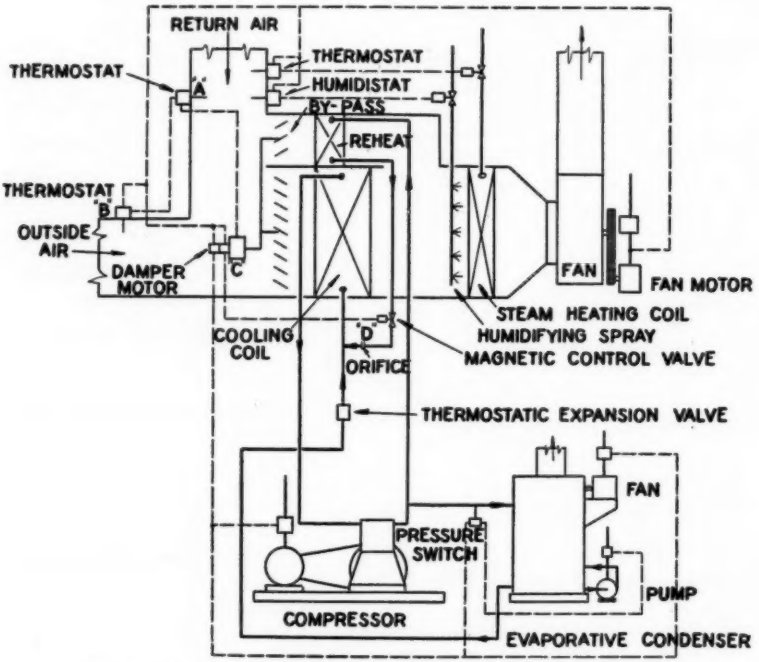

FIG. 9. YEAR 'ROUND INSTALLATION FOR USE OF PARTIAL REHEAT

damper. In this case the control is gradual. When, however, reheat is controlled by a magnetic valve as in Fig. 3 the regulation is of the *on* and *off* type. Both methods of control have been used successfully.

It will be apparent that there is an inherent difference between the regulation of *partial* and of *complete* reheat. With partial reheat a fall in dry-bulb temperature in the conditioned space below a permissible limit would be possible where the equipment operated for dehumidification at times of light sensible load. Some form of low temperature limit is therefore necessary. This low limit can stop the cooling before the dehumidification has reached a point that will satisfy the humidity control. On the other hand, this circumstance is impossible with complete reheat and no low limit device is used.

TYPICAL INSTALLATIONS

The application of refrigerant compressor gas reheat is illustrated by several typical installations which are described in detail. Fig. 7 depicts a comfort installation for summer cooling only incorporating complete reheat, and also shows there is no air bypass and the reheating coil is located beyond the cooling coil. There are two thermostats and a humidistat. The compressor may be operated by either humidistat *A* or thermostat *B*, these instruments being in parallel. A second thermostat *C*, set somewhat lower than *A* operates the magnetic valve *D*. If the dry-bulb temperature falls because the compressor is operating for dehumidification thermostat *C* will open valve *D*, thereby furnishing reheat. The compressor will continue to run and the reheat will cycle *on* and *off* until the humidistat is satisfied. The condenser is water cooled and the water regulating valve reduces the water flow so as to maintain the condenser pressure when the reheat is on.

There is shown in Fig. 8 a year 'round installation in which the reheat coil is divided into two steps so that the effect will approach gradual control more nearly than if a single reheat coil were used. With both reheat coils in operation the sensible cooling is entirely counteracted so that this is another example of complete reheating. As shown in Fig. 8 the reheating coils are located in the air stream downstream from the cooling coil and are followed by the humidifying spray and steam heating coil. It is worthy of note that, in this case, fixed orifices are used in place of a float drainer in the liquid outlets from the reheating coils.

For summer control there is a two step thermostat *A* and a humidistat *B* in the return air. The thermostat makes two contacts in succession with fall of return air temperature. As long as the temperature is so high that neither contact is made the compressor will run for sensible cooling. Should the temperature fall until the first contact is closed a normally closed relay *C* in the compressor starting circuit is opened as is also the magnetic valve *D* controlling the first section of the reheat coil. If the humidistat *B*, which is in parallel with this relay, does not require further cooling for dehumidification the compressor will stop. Should latent cooling be necessary the humidistat will keep the compressor running with the first section of reheat coil on. A further fall in dry-bulb temperature will cause the thermostat to close its second circuit and open the controlling magnetic valve *E* on the second reheat coil. Operation will then continue without further fall of dry-bulb temperature until the humidistat is satisfied and the compressor is stopped. The condenser is water-cooled and the water regulating valve serves to maintain the condensing pressure when the reheat is on.

For winter operation there is a thermostat operating a control valve in the steam supply to the heating coil. A humidistat operates the humidifying spray.

Fig. 9 illustrates the use of partial reheat with the heating coil in the air bypass and with the reheating capacity regulated by the bypass damper. This is a year 'round installation. As will be seen from Fig. 9 this installation incorporates an evaporative condenser.

For summer control there is a thermostat *A* whose setting is adjusted by an outside instrument *B* so as to call for a variation of inside with outside temperature changes. Thermostat *A* operates a damper motor *C* which gradually closes the coil face damper and opens the bypass, with fall of temperature,

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and vice versa. There is mounted on the damper motor shaft a cam which causes the magnetic control valve *D* to open somewhat after the bypass damper has left its closed position. This places the reheat coil in operation. The position of the bypass damper regulates the degree of reheat. The damper motor incorporates a second cam so arranged as to keep the compressor and evaporative condenser in operation at all times except when the bypass damper is wide open. It will be recognized that there is no definite summer humidity control in this installation. The reheat coil serves only to keep the compressor in operation longer, before the lowest permissible temperature is reached, than would be the case in a bypass installation without reheat. The effect of this is to produce lower relative humidities at light sensible loads than would be possible with the bypass alone. A pressure switch operates the circulating pump on the evaporative condenser intermittently to maintain the condenser pressure where the reheat coil is on.

For winter control a thermostat regulates the steam supply to the heating coil and humidistat operates the humidifying sprays.

Although there have been a good many variations in the arrangement of equipment and control the installations illustrated in Figs. 7, 8 and 9 will serve to show the principles involved in the application of compressor discharge gas reheat.

CONCLUSIONS

The provision of suitable means for regulating the capacity of air conditioning equipment and particularly for altering the relative proportions of sensible and latent heat removal, is of great importance. There are a number of ways to accomplish this with spray or coil conditioning apparatus but all of them, with the exception of reheat, are quite limited in the degree to which they can obtain relatively high latent cooling at times of small sensible load. By proper use of reheat it is possible to dehumidify with any degree of sensible cooling from the maximum provided for in the equipment down to zero. A source of reheat inherent in the operating cycle of a refrigerating machine is the compressor discharge gas. There is always more than enough heat automatically available from this source to neutralize all of the sensible cooling effect when dehumidification alone is required. Depending on the requirements of individual cases, compressor discharge gas reheat may be used to the extent of completely or only partially offsetting sensible cooling. In the latter case dehumidification without sensible cooling will not be possible but the proportion of latent heat abstraction will be greater than attainable without the reheat.

A sufficient number of installations of various types and sizes have been in operation for several years to demonstrate the practicability and effectiveness of compressor discharge gas reheat.

No. 1180

HEAT GAIN THROUGH WESTERN WINDOWS WITH AND WITHOUT SHADING

By F. C. Houghten * and David Shore,** Pittsburgh, Pa.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

HEAT gain through windows as a result of solar radiation is recognized as an important factor in summer cooling and air conditioning. The magnitude of this cooling load for unshaded and shaded windows, with different sun-arresting appurtenances, was reported [1] by the Laboratory for southern windows in 1934. The increasing interest in heat gain through various types of windows, including glass block, led to the building of two new cubicles or calorimeters on the roof of the U. S. Bureau of Mines Warehouse Building, Pittsburgh, in 1939. This new equipment was built on a turntable so that it could be made to face any horizontal direction, making it possible to study east and west windows.

The Technical Advisory Committee on Air Conditioning Requirements of Glass [2] provided a budget for the present study, the plans for which included the window and window shading arrangements listed in Table 1. The weather conditions during the past summer greatly limited the number of satisfactory tests which could be made. Nevertheless it is believed that the findings are of sufficient practical value to warrant their publication in this preliminary report pending a continuation of the investigation next year.

The two cubicles made it possible to simultaneously test and compare two window arrangements. The collection and analyses of the data were so arranged as to give the hourly total heat gain rate for the window, that gained as a result of radiation, and that gained by air to air transmission.

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[1] ASHVE Research Report No. 975—Studies of Solar Radiation Through Bare and Shaded Windows, by F. C. Houghten, Carl Gutberlet and J. L. Blackshaw. (ASHVE Transactions, Vol. 40, 1934.)

[2] ASHVE Technical Advisory Committee on Air Conditioning Requirements of Glass: M. L. Carr, *Chairman;* C. M. Ashley, L. T. Avery, F. L. Bishop, W. A. Danielson, H. C. Dickinson, J. D. Edwards, J. E. Frazier, S. O. Hall, E. H. Hobbie, C. L. Kribs, Jr., Axel Marin, R. A. Miller, F. W. Parkinson, J. H. Plummer, W. C. Randall, L. T. Sherwood, J. T. Staples, G. B. Watkins, F. C. Weinert.

Presented at the 47th Annual Meeting of the American Society of Heating and Ventilating Engineers, Kansas City, Mo., January, 1941.

as well as the percentage which the radiation gain was of the incident solar radiation on the outside of the glass. The comparative tests of two windows on single days permit a very close comparison, while the percentage which

TABLE 1—WINDOW ARRANGEMENTS STUDIED

NUMBER OF TEST	DATE	DIRECTION FACED	ROOM NUMBER	ACTUAL GLASS AREA SQ FT	CONDITION TESTED
3	7/25	West	1[a]	24	Unshaded single light window
		West	2[b]	24	Unshaded single light window
5	7/31	West	1	24	Unshaded single light window
		West	2	24	Completely shaded single light window
7	8/2	West	1	24	Window shaded on outside with awning
		West	2	24	Window shaded on outside with venetian blind
13	8/11	South	1	21.8	Unshaded twelve light wood sash window
		South	2	24	Unshaded one light wood sash window
17	8/17	West	1	24	Window shaded on outside by "shading screen" and on the inside by venetian blind
		West	2	24	Window shaded on outside by "shading screen"
37	9/18	West	1	24	Window shaded with venetian blind on the inside
		West	2	24	Window shaded with half-drawn roller shade inside
39	9/20	West	1	24	Unshaded single light window of heat absorbing glass
		West	2	24	Unshaded single light window of plain 3/16 in. sheet glass
47	9/28	West	1	21.8	Unshaded twelve light wood sash window
		West	2	21.1	Unshaded twelve light steel sash window
	10/4	West	1	24	Double glazed window—single light both outside and inside plain 3/16 in. sheet glass
50		West	2	24	Double glazed window—single light, outside glass heat absorbing, inside plain 3/16 in. sheet glass

[a] Room 1—South side with window facing west.
[b] Room 2—North side with window facing west.

the solar radiation gain on any day was of the incident radiation allows a less accurate, though satisfactory, comparison of any of the arrangements tested. The shading devices included a typical green canvas awning extending 4½ in.

on each side and to within 2 ft x 7 in. of the bottom of the window. At its lowest point it extended out horizontally a distance of 3 ft and was ventilated at the edges to allow the escape of trapped warm air. A typical dark green, cloth roller shade, 4 ft x 3 in. wide and opaque to any visible light radiation, was placed on the window frame so that the plane of the drawn shade was 1½ in. from the glass. The inside venetian blinds, completely covering the window opening, had bright aluminum curved slats, 2 in. in width, spaced 1⅝ in. apart, and so placed that their main axes formed an angle of 45 deg

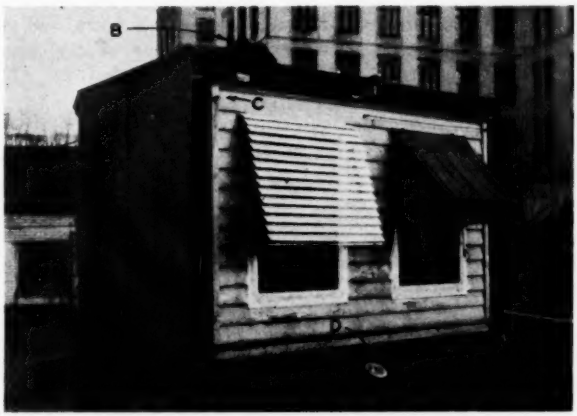

FIG. 1. TEST HOUSE EQUIPPED WITH AN OUTSIDE VENETIAN BLIND AND AN AWNING

A—Canvas shading screen
B—Cup anemometer
C—Pyrheliometer giving sun intensity on south wall
D—Pyrheliometer giving sun intensity perpendicular to rays

with the horizontal. The outside venetian blind was of cream painted metal slats and adjustable both as regards the angle of the slats and the slope of the blind away from the window; that is, its bottom could be extended away from the window as an awning. The shading screen was similar to a venetian blind, having very small slats, and designed to serve both the purpose of a fly screen and sun arrester. It had dark colored metal slats 0.050 in. wide, spaced 0.063 in. apart, and placed at an angle of 17 deg with the horizontal.

A photograph of the cubicles with a venetian blind on a western exposed, plain, single light, double-glazed window and a single light, double-glazed window shaded by an awning with heat absorbing glass on the outside and plain glass on the inside is shown in Fig. 1. A plan of the cubicles showing the arrangement of the test equipment is given in Fig. 2.

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TEST PROCEDURE

The test procedure, the data collected, and the analyses of the same in order to arrive at the heat gain through the glass by radiation and by transmittance were in most instances the same as in last year's study of heat gain through glass blocks,[3] the report of which cites these arrangements in greater detail than will be reported here. The cubicles were cooled by ice melting unit coolers. Ice meltage collected at 20 min intervals gave a measure of the total heat extracted in order to maintain the desired temperature level in the cubicle, usually 78 F. In all of the tests calibrated heat meters made it possible to determine the heat gain through the ceilings, floors, and walls other than the glass. Infiltration tests gave the magnitude of the heat gain by

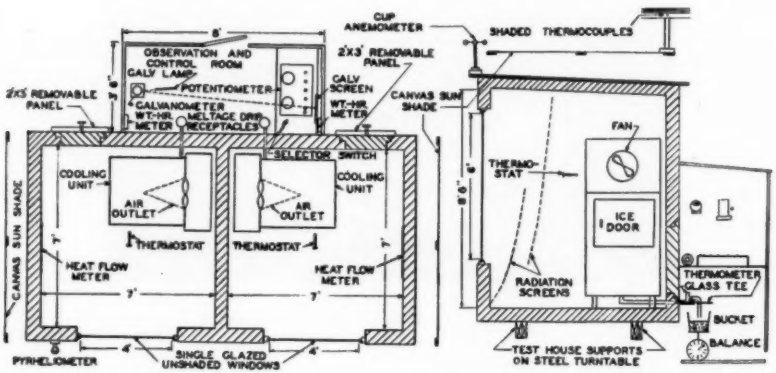

FIG. 2. TEST SET-UP FOR DETERMINING SOLAR RADIATION THROUGH WINDOWS

this avenue. The difference between the heat extracted and that gained through avenues other than the glass gave the total heat gain through the window. Calculations using the inside and outside air temperatures at a distance of 6 in. from the glass and the transmittance coefficient gave the heat gain through the windows resulting from the higher air temperature outside.

By difference the heat gain through the glass, resulting from radiation impinging on the outside, was determined. While it is recognized that this method offered a limitation in accuracy because of the fact that errors due to the calculations of heat gain by transmission and those resulting from the measurement of total heat gain through the window were thereby combined in the radiation gain, this method was nevertheless used because it offered

[3] ASHVE RESEARCH REPORT No. 1147—Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghten, David Shore, H. T. Olson, and Burt Gunst. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

the only possibility of determining the total effect which radiation had on heat gain to the room. If radiation had been measured by a pyrheliometer on the inside of the room it naturally would not have included the heat gain resulting from radiation but entering by conduction through the inside surface after being absorbed by the glass. In no case was the resulting heat gain from the inside-outside air temperature difference large, and, therefore, errors in the radiation values given, resulting from the difference, should not be significant.

It should be noted that not all of the heat gain charged to radiation necessarily passed through the inside surface of the glass as radiation, for some of the heat resulting from absorption of radiation in the glass itself naturally

TABLE 2—TEST DATA AUGUST 17 WEST WINDOW WITH OUTSIDE SHADING SCREEN. PLAIN 3/16 IN. SHEET GLASS AND WOOD SASH

SUN TIME	SOLAR INTENSITY ON WINDOW BTU/SQ FT/HR		OUTSIDE AIR TEMP. F	INSIDE GLASS TEMP. F	HEAT GAIN THROUGH GLASS						
									DESIGN		
	Observed	Design Aug. 1			Observed Rad. and Transmission	Calculated Air to Transmission	Radiation by Difference	Radiation in Per Cent Solar Intensity	Radiation Aug. 1	Transmission 95 Deg-Day	Radiation and Trans. Aug. 1
9–10	21.7	22	78.8	82.0	2.1	0.8	1.3	6.0	1.3	3.0	4.3
10–11	26.7	24	80.3	84.0	4.5	2.3	2.2	8.2	2.0	6 2	8.2
11–12	35.5	25	83.1	86.0	9.0	5.1	3.9	11.0	2.7	9.8	12.5
12–1	60.4	60	85.2	88.0	14.6	7.2	7.4	12.2	7.3	13.0	20.3
1–2	78.3	123	85.4	87.6	19.2	7.4	11.8	15.0	18.5	15.5	34.0
2–3	84.3	180	84.1	88.3	21.4	6.1	15.3	18.2	32.8	18.0	50.8
3–4	114.2	206	83.5	90.8	32.4	5.5	26.9	23.5	48.4	18.0	66.4
4–5	157.2	186	83.7	90.5	55.3	5.7	49.6	31.5	58.6	17.5	76.1
5–6	91.1	120	82.6	86.0	44.0	4.6	39.4	43.3	52.0	15.0	67.0
6–7	9.1	40	80.3	80.4	8.6	2.3	6.3	69.2	27.7	12.2	39.9
7–8	0	1.5	74.7	76.8	− 3.3	− 3.3	0	0	0	9.5	9.5

entered the room by conduction through the inside surface of the glass. This gain, however, all resulted directly or indirectly from the radiation impinging against the outside.

An Eppley pyrheliometer gave the total solar and sky radiation impinging against the plane of the window. Thermocouples gave the outside air temperature 6 in. away from the glass, the outside-inside glass surface temperatures, and the inside air temperature. Air velocity was observed by a cup anemometer whose rotations were indicated and read from a mechanical counter at one-half hour intervals.

In conducting a test, the coolers were charged with ice early in the morning and the rooms were closed. The desired inside temperature was maintained by manually adjusting the ice cooling rate to a value a little greater than that required and then adding a small amount of thermostatically controlled elec-

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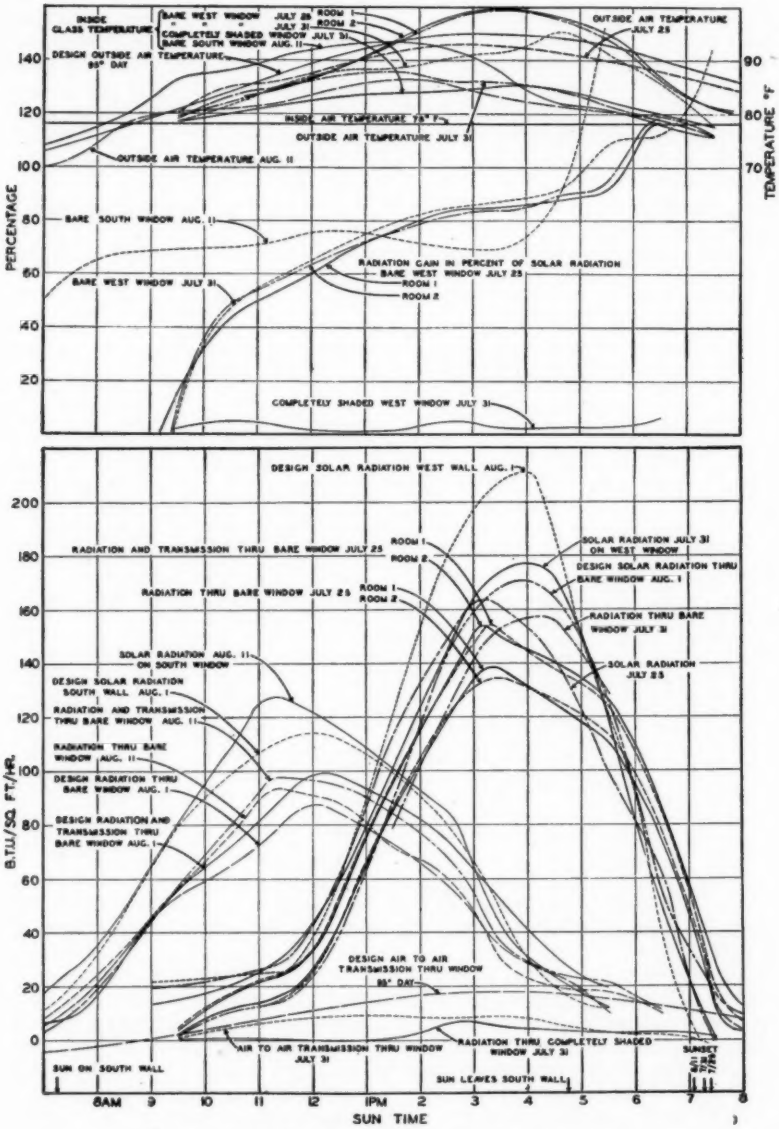
INSIDE GLASS TEMPERATURE
BARE WEST WINDOW JULY 25 ROOM 1
" " JULY 31 ROOM 2
COMPLETELY SHADED WINDOW JULY 31
BARE SOUTH WINDOW AUG. 11
OUTSIDE AIR TEMPERATURE JULY 25
DESIGN OUTSIDE AIR TEMPERATURE 95° DAY
INSIDE AIR TEMPERATURE 75° F.
OUTSIDE AIR TEMPERATURE JULY 31
OUTSIDE AIR TEMPERATURE AUG. 11
BARE SOUTH WINDOW AUG. 11
RADIATION GAIN IN PERCENT OF SOLAR RADIATION
BARE WEST WINDOW JULY 25
ROOM 1
ROOM 2
BARE WEST WINDOW JULY 31
COMPLETELY SHADED WEST WINDOW JULY 31
PERCENTAGE
TEMPERATURE °F
DESIGN SOLAR RADIATION WEST WALL AUG. 1
RADIATION AND TRANSMISSION THRU BARE WINDOW JULY 25
ROOM 1
ROOM 2
SOLAR RADIATION JULY 31 ON WEST WINDOW
DESIGN SOLAR RADIATION THRU BARE WINDOW AUG. 1
RADIATION THRU BARE WINDOW JULY 25
ROOM 1
ROOM 2
RADIATION THRU BARE WINDOW JULY 31
SOLAR RADIATION AUG. 11 ON SOUTH WINDOW
SOLAR RADIATION JULY 25
DESIGN SOLAR RADIATION SOUTH WALL AUG. 1
RADIATION AND TRANSMISSION THRU BARE WINDOW AUG. 11
RADIATION THRU BARE WINDOW AUG. 11
DESIGN RADIATION THRU BARE WINDOW AUG. 1
DESIGN RADIATION AND TRANSMISSION THRU BARE WINDOW AUG. 1
B.T.U./SQ. FT./HR.
DESIGN AIR TO AIR TRANSMISSION THRU WINDOW 95° DAY
AIR TO AIR TRANSMISSION THRU WINDOW JULY 31
RADIATION THRU COMPLETELY SHADED WINDOW JULY 31
SUNSET
SUN ON SOUTH WALL
SUN LEAVES SOUTH WALL
8AM 9 10 11 12 1PM 2 3 4 5 6 7 8
SUN TIME

FIG. 3. SOLAR RADIATION HEAT GAIN RELATIONSHIPS

Two, west, unshaded, single-light, single-glazed windows—July 25
One, west, unshaded, single-light, single-glazed window—July 31
One, west, completely shaded, single-light, single-glazed window—July 31
One, south, unshaded, single-light, single-glazed window—August 11

trical heat. Constant conditions were maintained for a sufficient length of time before the collection of heat gain data to insure uniform temperature conditions. In all tests, whether for a southern or a western exposure, the data were collected from early morning, thus obtaining a measure of the effect of sky radiation until the sun began to strike the glass itself. These tests were continued for a time after sundown until the heat ceased or became negligible.

TEST RESULTS

The results of a single test on August 17 on one western exposed window equipped with *shading screen* are given in Table 2. Similar results of the one most satisfactory day for each of the thirteen conditions listed in Table 1 are plotted in Figs. 3 to 9, giving the actual solar radiation intensity impinging against the outside of the window in Btu per square foot per hour, the total heat gain charged to each square foot of the glass, the calculated air to air transmission through the glass, and the heat gain through the inside surface of the galss resulting from radiation impinging against the outside. In some cases it was found desirable to omit either the total heat gain or the transmission gain curves in order to avoid confusion. The fact that the radiant gain is equal to the total gain by radiation and transmission, less the transmission gain, makes it possible to determine the magnitude of one in terms of the other two.

For comparison the design solar radiation intensity which would impinge against the outside of the window and the resulting radiation gain through the window based on the percentage relationship of actual radiation against the outside of the window and the radiation gain through the glass are given. In the top of the figures are also given the outside glass surface temperature, the design outside air temperature curve for a 95 F day, and in most cases the inside air temperature. The percentages which the radiation gains were of the total radiation intensities impinging against the outside of the windows are also given.

The testing program was started early in July and continued until completed on October 4. Tests on a given set-up were started every morning which gave promise of satisfactory weather conditions and were continued throughout the day unless the weather became unsatisfactory. This program was continued on each window until at least one test run of a full day in length was obtained. The very unsatisfactory weather with the small percentage of clear, sunny days accounts for the extended length of the program and for the acceptance of certain tests on days when the solar radiation was not very near design conditions.

DISCUSSION OF RESULTS

Considerable importance is attached to the design solar radiation and the design radiation and transmission gain through the windows. The term *design* is used in accordance with its usage by heating, ventilating, and air conditioning engineers in estimating values required in design and is construed as the outside solar radiation intensity on a summer day for which an engineer would design a cooling load. It naturally should not include the most severe day affecting the cooling capacity of the plant ever recorded for this would require

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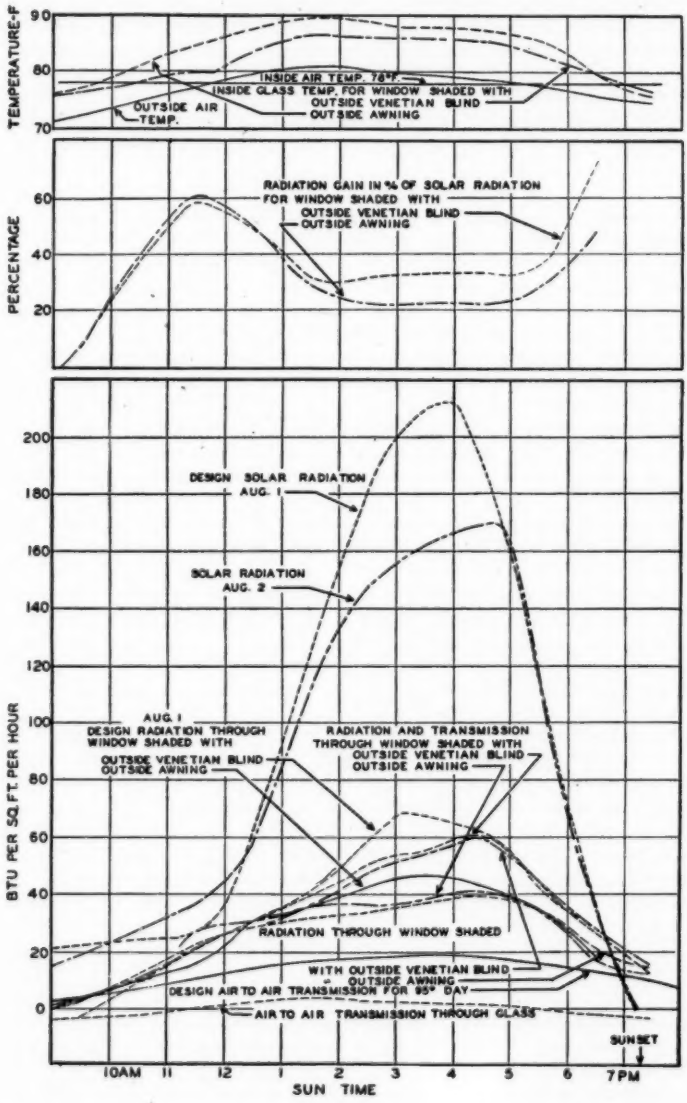


FIG. 4. SOLAR RADIATION HEAT GAIN RELATIONSHIPS FOR WEST EXPOSED WINDOWS (ONE SHADED WITH AN OUTSIDE AWNING, AND ONE SHADED WITH AN OUTSIDE VENETIAN BLIND)

an excessively large cooling system; but on the other hand it should be high enough so that a cooling plant calculated to have sufficient capacity for this assumed condition would adequately take care of all but a very few of the most severe days. Similarly, the term *design radiation* implies the radiation which would come into an occupied space on a day when the solar radiation intensity was that accepted for design purposes; and *design transmission gain* is the heat gain by transmission, unaffected by radiation, on a day when the inside temperature was at the point required for comfort on a design temperature day. It will be observed that, while the highest outside air temperature in the shade ever recorded for Pittsburgh was about 104 F, cooling design in Pittsburgh is usually based upon a day when the maximum reaches 95 F.

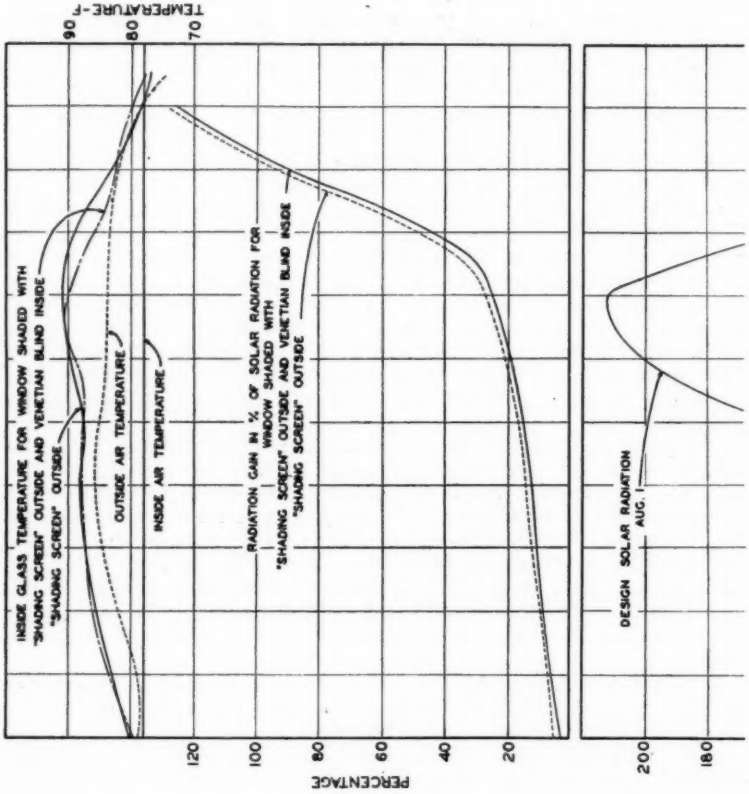
Design solar radiation intensity for August 1 was obtained by an analysis of weather bureau data in an earlier Laboratory report [4] and represents a fairly high solar radiation intensity such as may be expected to occur on two or three days during any summer. Design radiation gain is the gain which would have occurred on any given day for any window condition if the radiation intensity had been equal to the design solar radiation on August 1. This was obtained by applying the same percentage to the design solar radiation as the observed radiation gain through the window was of the actual solar radiation for the given time of test. Design air to air transmission through the glass is based upon the temperature maintained indoors and the outside air temperature cycle expected to occur on a day when the maximum air temperature reached 95 F. This transmission gain is applied to the design curves for August 1, but obviously the same transmission gain would occur on any 95 F day.

The preliminary tests of the two identical, unshaded windows in the two cubicles on July 25 were for the purpose of checking the similarity of results from the two cubicles. It will be noted from the curves in Fig. 3 that the comparison is very satisfactory. The comparison of the curves in this figure for an unshaded, west window on July 25 and for an unshaded, south window on August 11, serves to show the relation between the magnitude and the time of maximum heat gain through the western and southern exposed windows. This relation seems to verify the claims of designing and operating engineers that worse conditions are frequently had for a western than for a southern window.

The characteristics of the curves for the percentages which the radiant gains were of the total impinging radiation intensities against the outside of the window are of interest. The high values of the percentage rates, which occasionally became as high as 100 per cent, are accounted for by two facts. *First*, the accumulation of heat within the glass due to the absorption of radiation and the resulting rise in temperature is finally given up, partly to the inside, at a time when the solar radiation reached a low value or had ceased entirely. *Second*, the fact that the intensity of the solar radiation impinging against the outside of the glass became low due to the small angle of impingement, or low solar intensity, when a relatively small and frequently unimportant heat gain became a high percentage. It is obvious that for such conditions the accuracy of the percentage curves is low. However, on these occasions the total heat gain is low and therefore the percentage is not important.

[4] Loc. Cit. Note 3.

By shading the window *complete shading effect* was had, so as to entirely eliminate solar radiation impingement thereon and was obtained by arranging two thicknesses of canvas, sloping outward and downward before the window, so that the one nearest the window made an angle of approximately 10 deg with the vertical and the one farthest away twice this amount. It will be



noted in Fig. 3 that the heat gain charged to radiation through the completely shaded window was negligible. The results for the unshaded and completely shaded windows serve to give two reference points for comparison with the other commercially shaded windows. These curves show that the unshaded window admitted a little under 90 per cent of the radiation impinging against the outside when the solar radiation was a maximum, while the completely shaded window admitted only 2 per cent.

The radiation gains through the *shading screen* with inside venetian blind on August 17, show increased gains in the afternoon, while the curve indicates that the sun did not shine through between the slats until about 4:20 p.m. Computation shows that this change should have occurred at about 3:40 p.m., indicating a 40 min lag.

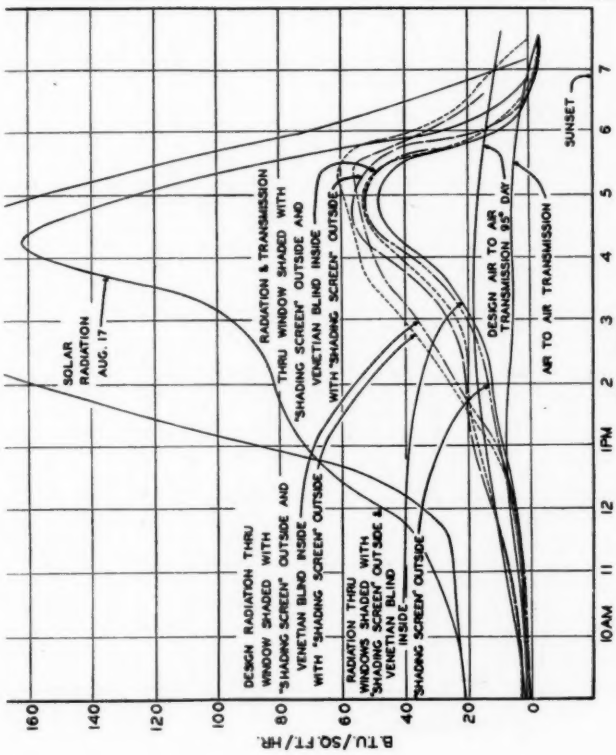
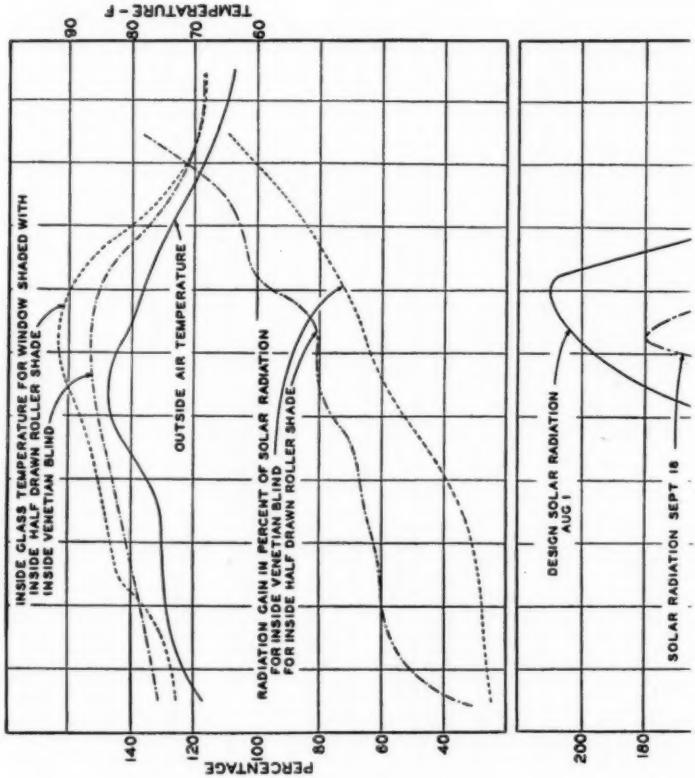


FIG. 5. SOLAR RADIATION HEAT GAIN RELATIONSHIPS FOR WEST EXPOSED WINDOWS (ONE SHADED WITH *Shading Screen* ON OUTSIDE AND A VENETIAN BLIND ON INSIDE; ONE WITH *Shading Screen* ON OUTSIDE)

Design data for heat gain through the different windows by radiation as plotted in the figures are given in Table 3. The first and last columns give the design solar radiation impinging on western exposed and southern exposed windows, respectively, for August 1. The design heat gain charged to radiation is expressed both in percentage of the design solar radiation intensity on the outside of the window, and in Btu per square foot per hour for each of the windows studied. These values are not always based upon the single tests

plotted in Figs. 3 to 9 but represent rather the average of these tests and others. Hence, while these values will not always agree exactly with the curves, they are given as representing more nearly the true averages of what might be expected under the circumstances. The total heat gain and that



charged to radiation for the metal and wood sash windows on September 28, Fig. 7 and Table 3, shows slightly higher values for the metal sash. These values were based upon the superficial glass area, including the area of the muntin bars. Since the muntin bar area on the metal frame window was somewhat less than that for the wood frame, this penalized the metal window. When corrected for this difference the radiation heat gain indicated as 64.0 and 69.0 Btu per square foot per hour for the wood and metal sash at 2:30 p.m. in Fig. 7 become substantially the same when based on net free glass area.

The design air to air transmission for a 95 F day is given in the table for a single-glazed window. These values may be added to the design heat gain due to radiation to give the total heat gain through any single-glazed window. If the total heat gain through double-glazed windows is desired, it

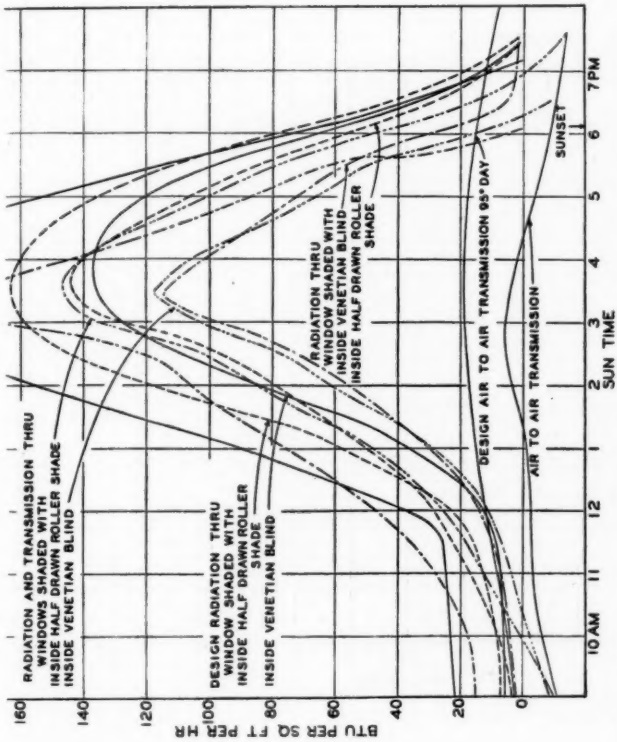
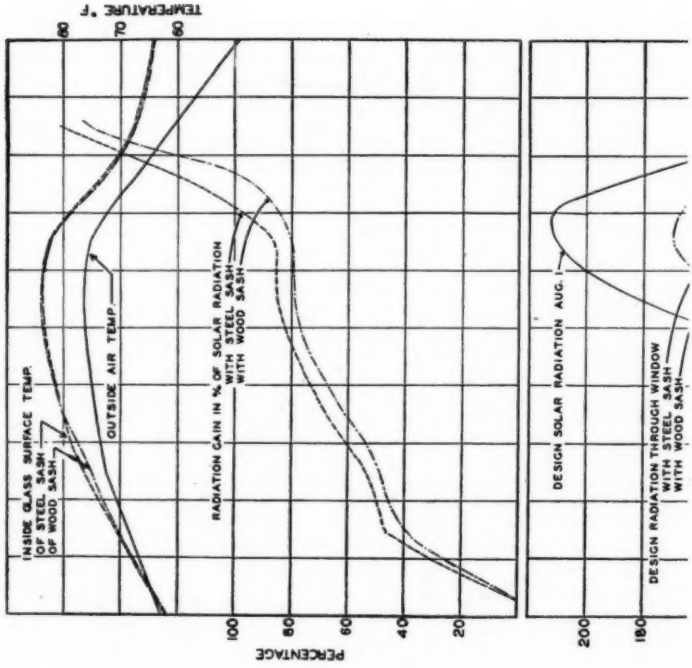


FIG. 6. SOLAR RADIATION HEAT GAIN RELATIONSHIPS WITH WEST EXPOSED WINDOWS (ONE SHADED WITH VENETIAN BLIND ON THE INSIDE; THE OTHER SHADED WITH A HALF DRAWN ROLLER SHADE ON INSIDE)

may be obtained by adding the design gain charged to radiation and the design transmission gain from the third column multiplied by 0.6.

While the result of this study adds considerably to the knowledge of heat gain through western windows, the few days during the past summer when the solar radiation even approached *design conditions* serve to emphasize that the results should be accepted with some reservation and that further study of some of the window arrangements is desirable. This is particularly true of the results for August 17, September 18, shown in Figs. 5 and 6,

when the observed solar radiation deviated considerably from *design conditions.* The results for western exposures before about 11 a.m., and following approximately one-half hour before sunset, are at times confusing, because of the small magnitude of the effects observed and the probable error in their measurement. This is particularly true for the percentage values given. It



is believed, however, that the results even for these periods serve some useful purpose as indications of probable values and trends. It should be emphasized, however, that the data for these periods are of no great value for design purposes and should be considered of academic, rather than practical value.

The curves frequently serve to emphasize the lag between the rate of the solar radiation effect on the outside of the window and the rate at which the heat was extracted by the melting ice. Examples are, the erratic percentages observed during the early hours, before the sun became effective

on the west wall, and around sunset, when the absorption of heat by the cooling system was still high and the sun effect was either low or had ceased to exist at all. These peculiarities give high percentages which are hard to justify until the low magnitude of the actual values entering into the percentages and the effect of the lag are considered. In this connection it

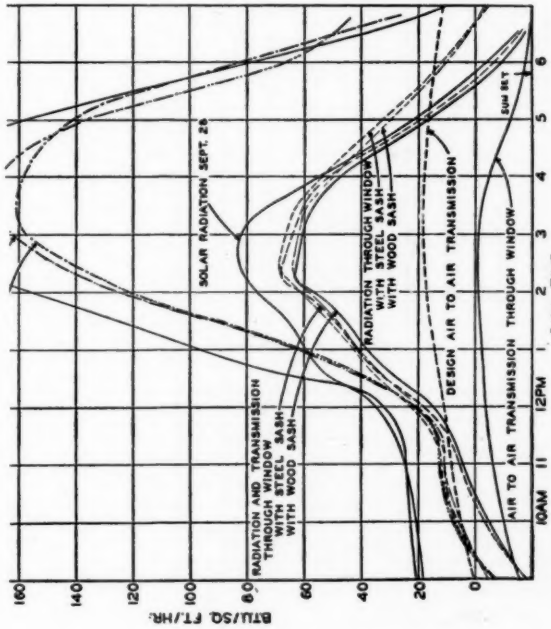


FIG. 7. SOLAR RADIATION HEAT GAIN RELATIONSHIPS FOR WEST EXPOSED WINDOWS, UNSHADED (ONE A 12-LIGHT, WOOD SASH, AND THE OTHER A 12-LIGHT, STEEL SASH WINDOW)

should be emphasized that unusual care was taken in this study to reduce this lag effect by installing the cloth drapes to absorb the solar radiation and quickly impart its heat to the air while in most practical applications this lag effect will be much greater. An example of this is shown by an office cooling study [5] made by the Laboratory, which indicated that a considerable per-

[5] ASHVE RESEARCH REPORT No. 1002—Cooling Requirements of Single Rooms in a Modern Office Building, by F. C. Houghten, Carl Gutberlet and Albert J. Wahl. (ASHVE TRANSACTIONS, Vol. 41, 1935.)

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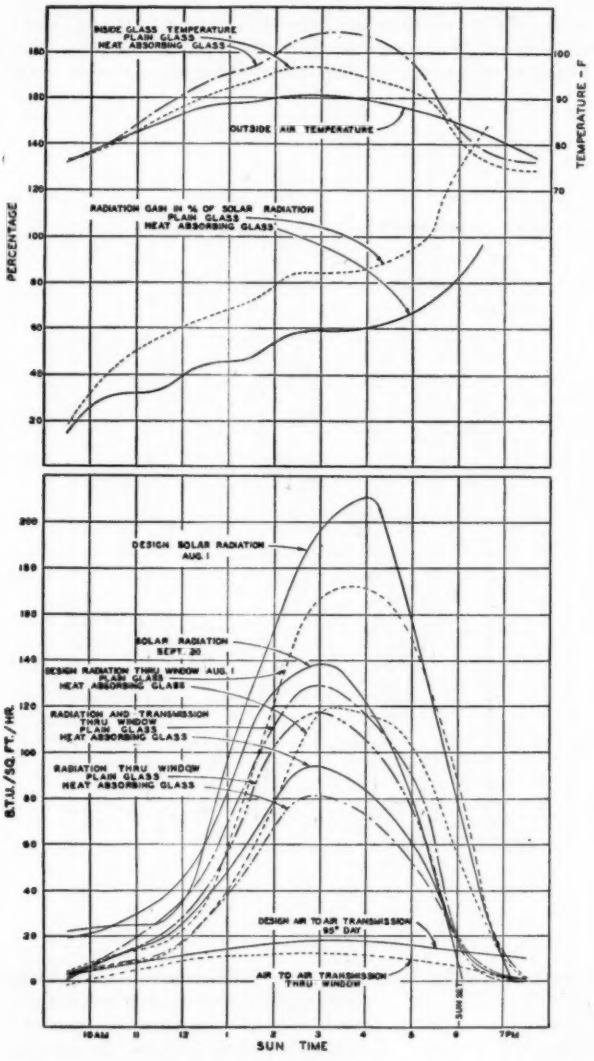

FIG. 8. SOLAR RADIATION HEAT GAIN RELATIONSHIPS FOR WEST, UNSHADED, SINGLE-LIGHT, SINGE-GLAZED, PLAIN AND HEAT ABSORBING GLASS

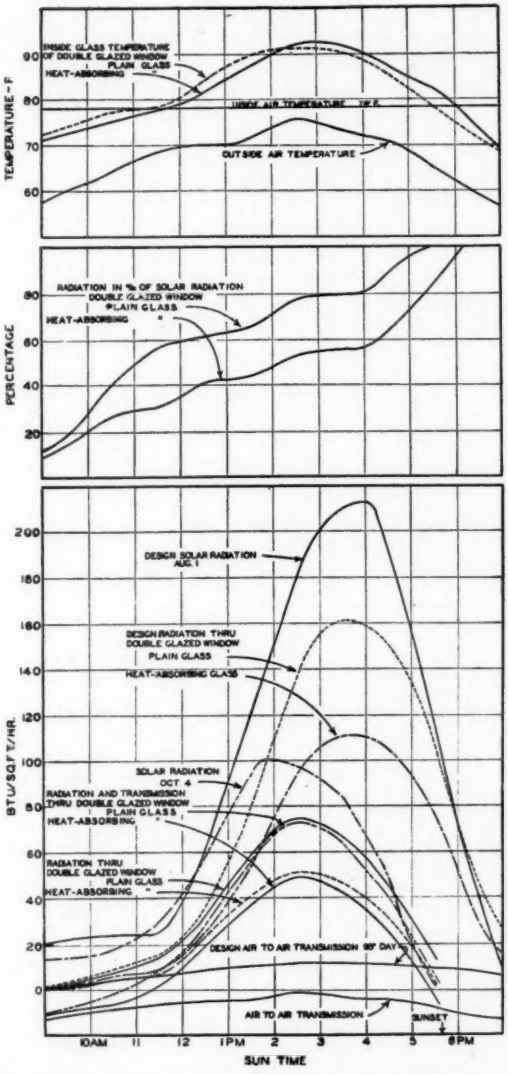

FIG. 9. SOLAR RADIATION HEAT GAIN RELATIONSHIPS FOR WEST, UNSHADED, DOUBLE-GLAZED WINDOWS (ONE GLAZED WITH PLAIN GLASS BOTH SIDES, THE OTHER GLAZED WITH PLAIN GLASS ON INSIDE AND HEAT ABSORBING GLASS ON THE OUTSIDE)

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TABLE 3—DESIGN HEAT GAIN DATA FOR SEVERAL WINDOW ARRANGEMENTS ON AUGUST 1, AT 40 DEG LATITUDE; PLAIN 3/16 IN. SHEET GLASS AND WOOD SASH, UNLESS OTHERWISE STATED

SUN TIME	SOLAR RADIATION AUGUST 1 40 DEG LAT. WEST WINDOW	AIR TO AIR TRANSMISSION 95 F DAY	DESIGN RADIATION GAIN IN BTU PER SQUARE FOOT PER HOUR AND IN PER CENT OF DESIGN SOLAR RADIATION INTENSITY ON WINDOW															
			WEST WINDOW															
			Unshaded One Light		Complete Shading		Outside Awning Dark Colored		Outside Venetian Blind[a] Light Colored		Shading Screen Outside Dark Colored		Shading Screen Outside Dark Colored Venetian Blind[b] Inside Aluminum Finish		Inside Venetian Blind[c] Aluminum Finish		Inside Roller Shade[d] Dark Green	
	Btu	Btu	Btu	Per Cent	Btu	Per Cent	Btu	Per Cent	Btu	Per Cent	Btu	Per Cent	Btu	Per Cent	Btu	Per Cent	Btu	Per Cent
5–6	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
6–7	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
7–8	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
8–9	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
9–10	22	3.0	4.0	18.3	0.44	2.0	1.7	7.7	1.9	8.8	1.3	6.0	0.9	4.1	5.3	24.0	6.8	30.7
10–11	24	6.2	10.5	43.7	1.20	5.0	9.4	39.4	9.0	37.5	2.0	8.2	1.7	7.1	6.3	26.4	10.0	41.7
11–12	25	9.8	15.5	62.1	0.55	2.2	15.1	60.5	14.6	58.3	2.7	11.0	2.4	9.6	7.1	28.2	15.1	60.4
12–1	60	13.0	39.8	66.4	1.14	1.9	29.8	49.7	29.3	48.8	7.3	12.2	6.6	11.0	20.7	34.5	38.8	64.6
1–2	123	15.0	88.4	71.9	2.00	1.6	34.8	28.3	39.1	31.5	18.5	15.0	17.1	13.9	55.3	45.0	83.9	68.2
2–3	180	18.0	153.0	85.0	8.82	4.9	41.2	22.9	57.2	31.8	32.8	18.2	29.3	16.3	106.2	59.0	144.4	80.2
3–4	206	18.0	175.5	85.2	5.97	2.9	46.8	22.7	67.6	32.8	48.4	23.5	44.7	21.7	136.0	66.0	166.0	80.6
4–5	186	17.5	165.9	89.2	4.70	2.5	41.3	22.2	62.1	33.4	58.6	31.5	53.4	28.7	133.9	72.0	153.5	82.5
5–6	120	15.0	123.0	102.5	3.24	2.7	34.3	28.6	43.4	36.2	52.0	43.3	45.4	37.8	98.4	82.0	115.2	96.0
6–7	40	12.2	43.9	109.7	2.48	6.2	19.3	48.3	28.9	72.2	27.7	69.2	27.2	68.1	36.4	91.0	46.0	115.0

[a]—So arranged as to keep sun off window.
[b]—Venetian blind slats at 45 deg.
[c]—Shade one-half drawn.
[d]—Double-glazed window—plain glass inside; heat absorbing glass outside.

TABLE 3—DESIGN HEAT GAIN DATA FOR SEVERAL WINDOW ARRANGEMENTS ON AUGUST 1, AT 40 DEG LATITUDE; PLAIN 3/16-IN. SHEET GLASS AND WOOD SASH, UNLESS OTHERWISE STATED—*Continued*

SUN TIME	SOLAR RADIATION AUGUST 1 40 DEG LAT. WEST WINDOW	AIR TO AIR TRANSMISSION 95 F DAY	DESIGN RADIATION GAIN IN BTU PER SQUARE FOOT PER HOUR AND IN PER CENT OF DESIGN SOLAR RADIATION INTENSITY ON WINDOW														SOLAR RADIATION AUGUST 1 40 DEG LAT. SOUTH WINDOW
									DOUBLE GLAZING				SOUTH WINDOW				
			Single Light Heat Absorbing Glass (Unshaded)		12-Light Wood Sash (Unshaded)		12-Light Steel Sash (Unshaded)		Two Sheets Plain Glass		One Sheet Plain Glass and One Sheet Heat Absorbing Glass[d]		12-Light Unshaded		1-LIGHT Unshaded		
	Btu	Btu	Btu	Per Cent	Btu	Per Cent	Btu	Per Cent	Btu	Per Cent	Btu	Per Cent	Btu	Per Cent	Btu	Per Cent	Btu
5–6	...	...	...	...	...	...	...	...	...	...	...	...	0.50	24.8	0.72	35.9	2.0
6–7	...	...	...	...	...	...	...	...	...	...	...	...	2.8	55.8	3.0	60.0	5.0
7–8	...	...	...	...	...	...	...	...	...	...	...	...	12.4	62.0	13.5	67.5	20.0
8–9	...	...	...	...	...	...	...	...	...	...	...	...	30.0	62.5	32.8	68.4	48.0
9–10	22	3.0	3.3	14.8	1.4	6.5	1.9	8.5	3.6	16.5	2.9	13.0	50.4	63.0	56.2	70.2	80.0
10–11	24	6.2	7.4	30.7	9.5	39.5	10.9	45.3	9.5	39.6	6.2	26.0	67.5	67.8	72.8	73.2	99.5
11–12	25	9.8	8.3	33.0	12.0	48.0	13.3	53.1	14.0	56.0	7.4	29.6	79.3	70.8	85.3	76.2	112.0
12–1	60	13.0	26.6	44.3	36.5	60.8	36.5	60.6	36.1	60.1	24.1	40.2	77.3	69.0	81.8	73.0	112.0
1–2	123	15.0	57.9	47.1	89.2	72.5	87.7	71.3	79.7	64.8	53.1	43.2	64.9	65.2	70.9	71.3	99.5
2–3	180	18.0	104.6	58.1	141.3	78.5	147.1	81.7	138.6	77.0	94.7	52.6	50.8	63.5	56.2	70.2	80.0
3–4	206	18.0	120.1	58.3	163.3	79.3	172.0	83.5	163.3	79.3	112.9	54.8	48.0	100.0	53.2	110.8	48.0
4–5	186	17.5	113.8	61.2	153.1	82.3	157.7	84.8	151.0	81.2	104.1	56.0	26.0	130.1	30.9	154.7	20.0
5–6	120	15.0	84.0	70.0	135.6	113.0	123.4	102.8	114.0	95.0	76.8	64.0	7.5	150.0	8.1	162.0	5.0
6–7	40	12.2	32.8	82.0	49.8	124.4	48.0	120.0	...	...	...	...	3.0	150.0	3.3	165.0	2.0

[a]—So arranged as to keep sun off window.
[b]—Venetian blind slats at 45 deg.
[c]—Shade one-half drawn.
[d]—Double-glazed window—plain glass inside; heat absorbing glass outside.

centage of the solar radiation entering the room during the afternoon was still in the concrete floor on which it fell at 5 p.m. In this connection attention is called to Fig. 4 showing from 18 to 20 Btu charged to solar radiation on the window, still being absorbed by the cooling system at 7:30 p.m., or after sunset, even though the outside air and inside glass surface temperatures were at the time lower than the inside air temperature.

DISCUSSION

L. T. AVERY (WRITTEN): Radiant energy is a fascinating subject. To the air conditioning engineer, the heat waves of radiation work with him in the heating cycle and against him in the cooling cycle. Little is known of low temperature radiation, although panel heating has been used and is popular in some kinds of work. Not too much is known about high temperature radiation. Any heat absorbing device, such as a venetian blind, special glass or sun screen, becomes a panel radiant heater. If this panel heater is in the room, this means that the effective temperature for comfort must be lower within its range, as compared with an interior location out of the range of window radiant heat.

The data here presented show quantitatively the difference in the heat reflecting or absorbing characteristics of the different materials inside and outside but do not show the difference in comfort effect due to hot walls. The temperature being held constant at 78 F, does not tell the whole story. An investigation should be made to determine variations in effective temperature for equal comfort near glass windows protected with these same materials.

The authors are to be complimented on getting so much data with so little available sunshine. Pittsburgh does not seem to be the best city in which to test solar radiation. Further work of this kind might be carried forward more effectively in Texas or some other location where there is a more dependable quantity and quality of sunshine. Investigations for cooling should be done when the outside air is 90 F or above. There seems too much chance for error when the room temperature is held at 78 F for example, and the test is run on days when the outside temperature is close to this level. Wall gains are easy to calculate, but somehow I feel we were on safer ground if the test on cooling was run on a hot day when the heat flow was very definitely from the outside in.

There is a need for further data on sun effect through glass block. A paper[6] was presented on this subject from tests run at the Research Laboratory, for glass block without sun protection. Because glass block has depth and mass, the influence of any inside or outside shade will be different from that on a window. The panel heating effect of glass block may influence comfort feeling. This study should, therefore, be extended to include venetian blinds, window shades and suncreen on the inside and outside of glass block.

Many engineers look at a paper of this kind and skip lightly over the salient features, so it is pertinent to mention here a few of the points of general interest.

1. Table 2, taken on August 17, was the only day a fair test was secured on the shading screen. Even on this day, the sun partially hid behind clouds from 1 p.m. to 4 p.m., to the extent that the observed radiation was considerably under the design. This may account for the fact that the apparent radiation transmitted through the screen is somewhat higher in percentage than shown in the manufacturer's tables, as prepared by Pittsburgh Testing Laboratory. This test should be repeated on a hot day with clear sunshine.

2. Fig. 5 shows a very slight decrease in the transmitted solar heat when the inside venetian blind is added to the outside shading screen. Even late in the afternoon,

[6] ASHVE RESEARCH REPORT No. 1147—Heat Gain Through Glass Blocks by Solar Radiation and Transmittance, by F. C. Houghlen, David Shore, H. T. Olson and Burt Gunst. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

when the sun would partially peek through the shading screen, the inside venetian blind saved an additional 3 per cent to 4 per cent only, not enough to be important.

3. Fig. 6 shows very plainly that the solar load passing through the inside venetian blind, fluctuates violently with the angle of the sun. Table 3 shows that the percentage transmitted, starts at 34.5 per cent at 12 to 1, and rises to 72 per cent at 4 to 5. This corrects the former erroneous idea that the inside venetian blind stopped 52 per cent and passed 48 per cent at all times.

4. Attention should be called to the change in values for solar radiation as compared with the tables previously used in THE GUIDE.

With the data in Table 3, the air conditioning engineer can directly figure the Btu per square foot for western exposure under any common condition of construction. This table is certainly a distinct contribution to the art. The authors are to be congratulated on this splendid piece of work. It is hoped that solar heat and its control will be further investigated to extend our knowledge of this important subject.

M. L. CARR (WRITTEN): This paper presents the first results obtained in an investigation that we hope funds can be found for continuing in the summer of 1941. The phases of the general subject which are reported upon today are those that the Technical Advisory Committee on Air Conditioning Requirements of Glass thought should be investigated first in a study of sun arresting appurtenances. Owing to the fact that in carrying on such investigations we are at the mercy of the weather, only a limited amount can be accomplished in the average Pittsburgh summer.

Not only are there other phases of the subject that the Committee believes should be investigated, but additional work along certain lines already investigated would be very desirable. Few research projects in our field, or any other field for that matter, yield results that should not be verified by additional work. It is hoped that the presentation of this paper will evoke discussion, criticisms and suggestions. Your committee desires such for consideration when it meets about the middle of March next to make plans for the coming sunshine season.

C. M. ASHLEY (WRITTEN): This paper is a real contribution to our knowledge of sunlight radiation through glass, both through the enlarging of the scope of the work previously done by bringing into play new instruments and new techniques.

Unfortunately, however, the weather conditions were very adverse during the summer of 1940 and due to this and possibly other causes many of the results must be accepted with considerable reservation. I would, therefore, suggest that this paper be viewed as a progress report subject to some modifications when more complete data are available.

Concerning the incident solar radiation, undoubtedly the new pyrheliometer represents a distinct advance over the older type in that it measures sky radiation as well as direct solar radiation. However, the results would be more readily acceptable if some information could be made available as to the check between the two. It seems to me that the radiation is possibly too high just before noon and too low in the late afternoon. Possibly the late afternoon condition is a characteristic of the Pittsburgh region and I believe some consideration should be given to checking the heliometer readings at other geographical locations.

I have some comments concerning the individual test results. The Btu values for awnings and outside venetian blinds seem to me to be too high. In this case there is some internal evidence that the heat gain through other walls of the cubicle are not fully deducted.

This shading screen tests seem to me to be among the most satisfactory despite the bad character of incident radiation curve.

The percentage of the heat gain for inside venetian blinds and inside roller shade seems too high in relation to previous tests. Possibly in the case of the shade this is accounted for by the color and fit.

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From theoretical consideration the percentage of heat gain for single heat absorbing glass seems to be too low, while that for the double glazing with heat absorbing glass apepars to be high.

It is to be hoped that the experimental program during the summer of 1941 will provide answers to these problems.

E. H. HOBBIE (WRITTEN): I feel that this paper is a very valuable contribution to the Society and I thoroughly appreciate the amount of work involved in obtaining the information presented.

However, I believe that additional research could be carried on which would make this report even more valuable. I should like to see figures included in Table 3 on a single-light plain glass, also a single light of heat absorbing glare reducing glass for comparison with the figures already given for a single light of heat absorbing glass unshaded. I should also like to see figures for double glazing with one sheet of plain glass on the inside and a sheet of heat absorbing glare reducing glass on the outside.

As this subject of heat gains through windows and the control of same by the use of awnings, venetian blinds and other methods of shading is so closely associated with daylight illumination, it is hoped that a joint committee of the ASHVE and the *IES* may some day be formed to give us the benefits of their studies.

W. C. RANDALL: These comments deal with Table 1, the curves given in Fig. 7, and Table 3 showing the solar radiation heat gain relationship for unshaded 12 light wood and steel windows.

Table 1 shows the glass area for wood and steel sash to be 1.8 and 1.1 sq ft. These figures should be 21.8 and 21.1 sq ft for wood and steel, respectively. I cannot understand this difference of 0.7 sq ft since it was intended that the net glass areas were to be identical.

To gain a true relationship between the wood and steel window values and other values in Table 3, it is necessary to base all computation on an equivalent area of 24 sq ft, which includes the area of the muntins and is more typical of windows, although still not the masonry opening normally considered the window opening.

The following table shows the design radiation gain in Btu's per square foot per hour, and radiation gain in per cent of design solar radiation based on the 24 sq ft; the same as other values in Table 3.

T. T. TUCKER: The authors state that this is a preliminary report pending a continuation of investigation next year. I would like to ask the authors if it is proposed to continue this investigation next year. Then, they further state the solar radiation was not very near design conditions. I would like to ask the authors if, in their opinion, we can use Table 3 for design conditions.

PROPOSED TABLE BASED ON 24 SQ FT GROSS GLASS AREA
(For Both Wood and Steel Windows, based on Fig. 7)

12-LIGHT WOOD		12-LIGHT STEEL	
Btu	Per Cent	Btu	Per Cent
1.3	5.8	1.6	7.2
8.5	35.4	9.5	39.6
10.8	43.2	11.5	46.0
33.2	55.3	32.1	53.5
81.0	65.8	77.4	62.8
128.1	71.3	129.5	71.9
146.0	71.9	149.0	73.4
139.0	74.7	139.0	74.7
104.4	87.0	108.2	90.2

MR. HOUGHTEN: In regard to the questions raised by Mr. Tucker, the question whether this study is continued during the summer of 1941 will depend upon budget arrangements made by the Committee on Research.

The second question was with regard to the use of these data for design purposes. I would say, yes, although the sunradiation never was always perfect; we feel we did a fairly good job in translating those data to August 1.

While this investigation comes quite far from giving all we would like to have obtained in this study, the results are better data than anything we have had before. As pointed out by Mr. Ashley, and as indicated in this paper, there should be further study to make the data more complete, but it will be my feeling that as of today, these data are better than anything we have.

J. H. CARTER: In the latter part of this paper there is a paragraph which indicates that radiation gains through shading screen with inside venetian blinds should have occurred at 3:40 p.m. on August 17, but that the curve of results shows that the sun did not shine between the slats until 4:20 p.m., indicating a 40-min heat lag. I find this statement rather puzzling, and should like to be enlightened upon it.

The delivery of radiant energy is practically an instantaneous effect. If an apparent lag of 40 min exists, it must be attributable either to errors in computation or observation, or to apparent errors caused by instrument lag but subject to calibration. Could the lag have been caused by the time required for the muslin to absorb the radiation?

There is a possibility which occurs to me which is somewhat far-fetched but perhaps of theoretical interest. When the sun's angle is such that it just starts to shine through the slats of the shading screen, the screen openings are of very small size, and the screen might be considered as an optical grating. The wavelength of infra-red or heat rays ranges from 8,000 up to 4,000,000 ångström units, or from $\frac{1}{30,000}$ up to about $\frac{1}{64}$ of an inch in length. Therefore, until the screen openings offer a slit of more than $\frac{1}{64}$ of an inch width for the sun's rays to pass through, I believe that we can look for a diffraction effect, which will diffuse the heat rays throughout the cubicle, instead of concentrating them upon the muslin target. It is possible that this optical effect may account for the apparent 40-min lag.

MR. HOUGHTEN: It is apparent in all of this work, that in spite of our aims and efforts to cut down lag as far as possible, that there is a lag from 20 to possibly as much as 40-min, although hardly that long between the time the maximum rate of heat gets through the glass until the time it is picked up by the cooling system. Perhaps the best indication of that is in curves for shading screen, where there is a definite knee, or turning point in the curve, which correlates fairly well with the time that the sun actually begins to peek through, between the slats.

The actual knee in those curves appears somewhere between 20 and 30 min later than the sun first starts to get through. So, this 40 min you speak of in this case, which would be a little hard to check up in a particular instance, is probably lag. But then, remember, when you come to apply these data in almost any installation, you might think you will have more lag, and you must take that into consideration in your application, some way.

R. A. MILLER: I want to raise the question, which I have raised two or three times before in the discussion of transmission of radiant heat, conductance of heat, and the application of the word *transmission* to the conductance of sensible heat, particularly as applies to glass.

If you confuse the terminology we are going to confuse the meanings of what we are trying to say. Radiant heat is never conducted, but always transmitted. Sensible heat may be transmitted, perhaps through hot water pipes, but not transmitted through glass.

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Then I believe the question was asked, as to whether the lag in any of these measurements might be due to the mass of the glass in the thicknesses in which we ordinarily use glass in glazing. The mass is not very significant as a heat storage medium. The losses heat very rapidly. It is a rather good conductor for reasonable amounts of heat, and it consequently lags in measurement due to radiant energy transmission, or sensible heat conductance through the glass will hardly ever be of any moment unless you take very radical measurements that do not allow for any kind of dissipation of heat.

G. L. TUVE: In regard to the comparison between venetian blind awnings, or outside venetian blinds, and canvas awnings, it happens that at Case School of Applied Science, Cleveland, we have two cubicles very much like yours, operated under about the same outdoor temperatures. From the beginning of July until October 1, 1940, we too tested about the same set of units that you used, but ours was a *hot attic method* instead of a refrigeration test.

For this demonstration, we compared two window treatments under identical conditions. We had an automatic recording system that switched over every 20 min from one house to the other. Our results were calculated from the temperature records on a chart, and we had the two rooms on the same chart.

I have made some averages from your data showing the Btu per hour transmitted by radiation. I find the wood or steel sash, double glass or roller shade are all practically the same. Call this 100 per cent; the inside venetian blind then transmits 80 per cent, the venetian blind or slat-type awning transmits 39 per cent, shading screen 32 per cent, and canvas awning 28 per cent. These percentages are based on your measurements between 2 p.m. and 6 p.m. only. I think this would be the important part of the day for the west wall.

My question is, do you have any explanation of the venetian blind awning giving 39 per cent heat transmission compared with only 28 per cent for the canvas awning? This means practically 40 per cent more heat admitted with a venetian blind awning than with canvas awning. Our tests showed very little difference between these two units.

MR. HOUGHTEN: I would not know how to account for the difference. Of course, the sun does get through the slats of venetian blinds, and not through the awning. A part of the answer may be in the type of outside venetian blind used. You notice this venetian blind shown on the house on page 2 is an exterior venetian blind. It is not a venetian blind such as used on the inside; but is the commercial type that stands out from the window.

PROFESSOR TUVE: We used exactly the same thing as far as I can determine. At what angle did you set the slats?

MR. HOUGHTEN: They were set at 40 deg, and at such an angle so as to keep out the largest percentage of sun radiation.

MR. MILLER: You get a lot of reflection from one vane to another, transmitted directly as heat.

MR. HOUGHTEN: Possibly it has an effect. I would not know without checking.

PROFESSOR TUVE: Ours were painted a dark color, yours were light; that appears to be the only difference.

MR. HOUGHTEN: You can get into a very good mathematical argument with yourself, as to which does the best. If black, the awning, itself, or venetian blind would get hot and be a source of radiation. On the other hand, it raises the temperature near the glass, impinges against the glass and raises the outside surface temperature.

PROFESSOR TUVE: That low temperature radiation will not go through the glass.

No. 1181

SEMI-ANNUAL MEETING, 1941

San Francisco, Calif.

SAN FRANCISCO was the scene of the 1941 Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the 52nd Anuual Convention of the *Heating, Piping and Air Conditioning Contractors National Association,* during the week of June 16. Because this was the first occasion on which either of the organizations had met on the Pacific Coast it was fitting to have the opening session as a joint function, after which the membership of both organizations attended the opening of the Pacific Heating and Air Conditioning Exposition.

Headquarters for the ASHVE was established at the Palace Hotel, and 360 members and guests registered. The St. Francis Hotel was the center of activity for the *HPACCNA* and their registration numbered 450.

The joint session of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *Heating, Piping and Air Conditioning Contractors National Association* was called to order by Walter L. Fleisher, president of the ASHVE, at 10:00 a.m., on Monday, June 16, in the Colonial Room of the St. Francis Hotel, San Francisco, Calif.

Major Harry A. Shuder, U. S. Army Chaplain gave the invocation and John Henderson sang "God Bless America." The audience then sang "America," led by Messrs. Henderson and N. H. Peterson.

Chairman Fleisher introduced the Honorable Paul Peek, Secretary of State of California, who gave a message of welcome on behalf of Governor Olson, and a brief response was made by J. E. McNevin, president of the *HPACCNA.* C. E. Bentley, president of Golden Gate Chapter of the ASHVE extended a welcome on behalf of the local chapter and Daniel Hayes, president of the *Heating, Piping and Air Conditioning Contractors of San Francisco,* assured the members assembled that it was a great pleasure for the local contractors to entertain the visiting members.

Mr. McNevin presented Frank E. Moran, president of the *Chicago Master Steamfitters Association.* Mr. Moran said that the Chicago association for many years had enjoyed a very conservative and well directed guiding hand centered in Louis Braun, who had just completed his 25th year as executive secretary. In commemoration of this occasion, he said, "the Chicago association desires to present a testimonial of appreciation, and, as president, I take great pleasure in presenting this diamond ring to Mr. Braun."

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Mr. Braun made a brief response and expressed his appreciation to the Chicago members for their fine tribute to his service.

Chairman Fleisher then called on G. M. Simonson, chairman of the ASHVE Committee on Arrangements, to give some announcements about the entertainment.

Chairman Fleisher introduced Mr. A. Appleton, president of the San Francisco Chapter of *American Institute of Architects,* who gave an entertaining discussion on the necessity for close collaboration between architects, engineers and contractors.

Announcement was made by Mr. McNevin of the appointment of the *HPACCNA* credentials committee.

Chairman Fleisher said that in his opinion the joint session should not close without having a few remarks from some of the men who have been instrumental in the building of both organizations. He then introduced Harry M. Hart, Chicago, and W. H. Driscoll, Syracuse, N. Y.

Dr. J. C. Geiger, director of Public Health for the city of San Francisco, was then introduced by Chairman Fleisher, and he presented the greetings of Mayor Angelo J. Rossi, and assured the assembly that San Francisco was very happy to entertain the group that is doing so much for the comfort of America. The session adjourned at 12:45 p.m.

The first technical session of the 1941 Semi-Annual Meeting was called to order on Tuesday, June 17, at 1:30 p.m., by W. L. Fleisher, president of the Society. He invited G. M. Simonson, general chairman of the Committee on Arrangements and past president of the Golden Gate Chapter, to outline the arrangements for the meeting.

Mr. Simonson welcomed the members and guests to California and announced that it was gratifying to learn that the registration had already far surpassed expectations. He assured the members that it would be a great pleasure for the Golden Gate Chapter and the committee to make their visit an enjoyable one.

President Fleisher acknowledged the Society's appreciation of Mr. Simonson's words of welcome, and proceeded with the technical papers.

Following the presentations of the various technical papers, president Fleisher announced that amendments to the By-Laws of the Society would be read by A. V. Hutchinson, secretary of the Society. These amendments were presented in accordance with the terms of Article B-XVI, in a form submitted by the Committee on Constitution and By-Laws as follows:

Amendments to the By-Laws

Article B-III—Section 7—When the Admission and Advancement Committee has acted favorably upon a Candidate's application and assigned his grade the Council shall vote upon the election of the proposed Candidate for membership by letter ballot.

To be Amended as follows: When the Admission and Advancement Committee has acted upon a candidate's application and recommended his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot.

Article B-IV—Section 1—The admission fee of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1942 and thereafter the admission fee of Members and Associate Members shall be fifteen dollars ($15.00); of Junior Members five dollars ($5.00); and of Student Members two dollars ($2.00). Admission fee must accompany application.

To be Amended as follows: The Admission fee of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1945 and thereafter the admission fee of Members and Associate Members shall be fifteen dollars ($15.00); of Junior Members five dollars ($5.00); and of Student Members two dollars ($2.00). Admission fee must accompany application.

Article B-IV—Section 2—The annual dues of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1942 and thereafter the dues of Members and Associate Members shall be twenty-five dollars ($25.00); of Junior Members twelve dollars ($12.00); and of Student Members five dollars ($5.00).

To be Amended as follows: The annual dues of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1945 and thereafter the dues of Members and Associate Members shall be twenty-five dollars ($25.00); of Junior Members twelve dollars ($12.00); and of Student Members five dollars ($5.00).

Article B-VIII—Section 7—There shall be an Admission and Advancement Committee of three (3) members, appointed by the Council, for a term of three (3) years, the term of one member expiring each year. This Committee shall receive from the secretary all applications for membership, make rigid inquiry as to the eligibility of candidates, and report to the Council only such as have been approved. In case of disapproval, only the proposers and the applicant shall be notified of such action. The proceedings of the Committee shall be private and confidential. It shall be the duty of this Committee each year to recommend to the Council, for transfer of grade of membership all Junior Members who have reached the age of thirty (30).

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The four amendments endorsed by the Council were voted upon separately and duly carried by unanimous vote.

President Fleisher before adjourning the session, introduced J. E. McNevin, president of the *Heating, Piping and Air Conditioning Contractors National Association,* holding its 52nd Annual Convention at the St. Francis Hotel during the same week. Mr. McNevin acknowledged the introduction and extended an invitation to all to attend the meetings at the St. Francis. Mr. McNevin was glad of the opportunity of being present and hoped to be able to get back to more of the technical sessions.

President Fleisher asked for further business and then announced that the session was adjourned at 4:15 p.m.

President Fleisher called the second technical session to order on Wednesday, June 18, at 9:30 a.m., and the technical papers scheduled were presented.

B. W. Farnes, Portland, Ore., extended a cordial invitation on behalf of the Oregon Chapter to the visiting members and their families to be their guests for a day or so on their return trips, to visit Bonneville Dam, Mount Hood, and take the Columbia River Highway trip. The trip included the large power plant at Bonneville Dam, the fish hatchery and the fish ladders which assist the salmon in getting up to the spawning ground, something

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which they felt the folks from the East were not familiar with and would be interested in seeing. Many of the members showed interest in Mr. Farnes' announcement, and President Fleisher was among those who hoped to enjoy the Oregon Chapter's hospitality.

For the benefit of the members returning via Los Angeles, Mr. Hutchinson announced that word had been received that the Southern California Chapter offered its services to those interested in visiting the movie studios of Warner Bros. or Twentieth-Century Fox. H. H. Douglas, past president of the Chapter, A. J. Hess, and Leo Hungerford, president and secretary of the Chapter, were ready to welcome the members as they arrived.

President Fleisher then proceeded with the technical program.

Professor Eastwood, first vice-president of the Society, then took the chair and called for discussion, from the floor on the technical papers.

Following several announcements concerning the next technical session the second technical session adjourned at 12:15 p.m.

The third technical session was called to order on Thursday, June 19, at 9:30 a.m., by President Fleisher, and a Resumé of Society Research in Co-operative Institutions, which was to have been delivered by A. E. Stacey, Jr., chairman of the Committee on Research, was presented by Mr. James. Twenty-four years ago the idea of fundamental research on heating, ventilating and air conditioning problems was suggested to the Society, as related by Mr. James, and in 1919 the ASHVE Research Laboratory, located in the U. S. Bureau of Mines, Pittsburgh, Pa., now under the direction of Dr. F. C. Houghten, became a reality. Since that time, he explained, over $600,000 has been spent on a variety of subjects of which approximately 50 per cent of that amount has been received from membership dues and the balance has been contributed by various manufacturers and trade associations.

Mr. James also stated that in addition to maintaining its own Research Laboratory in Pittsburgh, the Committee on Research at the present time is actively sponsoring cooperative research in 13 institutions. At the conclusion of Mr. James' presentation, President Fleisher expressed his gratitude to him for presenting the activities of the Society in such an excellent manner, and S. R. Lewis joined in thanking Mr. James and suggested that a message be sent to Mr. Stacey expressing a vote of thanks for the progress being accomplished by the Committee on Research under his direction.

M. F. Blankin, Philadelphia, Pa., treasurer of the Society and chairman of the Resolutions Committee, was then called upon and presented the following resolutions prepared by the Committee, which were voted upon and unanimously carried:

Resolutions

WHEREAS, the members of the ASHVE who are attending the First Pacific Coast Meeting of the Society desire to express their sincere appreciation for the splendid program which has been presented, now therefore,

BE IT RESOLVED, that an expression of thanks and appreciation be adopted and spread upon the minutes of the Society and copies thereof be transmitted to each of the following persons and agencies that have contributed to make this meeting so enjoyable for the members of the Society who attended:

To the Golden Gate Chapter of the Society and their ladies who have received us with such gracious hospitality;

Also to the three other Pacific Coast Chapters who have cooperated so well with the Golden Gate Chapter;

To the members of the Committee on Arrangements for the fine program of entertainment;

To the authors of the technical papers for their valuable contributions and discussions;

To Mayor Angelo J. Rossi for his assistance in officially opening the Exposition and the many courtesies which he extended to us;

To the Management and Staff of the Palace Hotel for their excellent service;

To Mr. Walter Swanson and the Staff of the San Francisco Convention and Tourist Bureau for their fine cooperation and assistance;

To the newspapers for their coverage of our sessions and to the trade papers for their excellent publicity and attendance at the meetings;

To the railroads and airlines for their courtesies and efficient service;

To the Postal Telegraph Company for its courtesy in furnishing the registration service board;

To the University of California for arranging for the inspection trip and meeting at the Engineering Building and the tea at International House;

To Dr. Chauncey D. Leake for his splendid talk at the Semi-Annual Banquet;

To Dr. B. M. Woods for the excellent job he did as Toastmaster at the Semi-Annual Banquet;

To Mr. C. F. Roth and his associates for arranging a splendid exposition and to the exhibitors for their participation and cooperation,

And finally to recognize A. V. Hutchinson and his eighteen years of service in the Society and his fifteen years of continuous service as Secretary.

Respectfully Submitted,

M. F. BLANKIN, *Chairman*,
M. W. BISHOP, R. L. BLANDING

Clyde E. Bentley responded to Mr. Blankin and expressed the appreciation of the Golden Gate Chapter for the opportunity of acting as host to the Society.

President Fleisher then announced that the last technical session would convene at the University of California, Berkeley, at 2:00 p.m. The session adjourned at 11:45 a.m.

On Thursday, June 19, President Fleisher called the final technical session to order at 2:00 p.m. in Room 104 of the Engineering Building, University of California.

ENTERTAINMENT

The Committee on Arrangements under the able leadership of G. M. Simonson had arranged an elaborate entertainment program for visiting members and everyone who attended the five-day meeting retains many pleasant memories of his visit with the Pacific Coast members.

The first get-together was the joint luncheon of engineers and contractors at the St. Francis Hotel on Monday, after which Presidents Fleisher and McNevin and Honorable Angelo J. Rossi, mayor of San Francisco, officially opened the Pacific Heating and Air Conditioning Exposition at the Civic Auditorium.

At 2:00 p.m., a golf tournament was held at the Lakeside Country Club and those who participated played the ocean course. The Research Cup was

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awarded to James Gayner, San Francisco, with a score of 86-15-71. Other awards were made to C. P. Hensley and Fred W. Kolb, San Francisco. In the competition for the Eichberg Memorial Cup the Golden Gate Chapter team had the best score; James Gayner, 86-15-71; W. P. Scott, Jr., 86-9-77; Fred W. Kolb, 102-24-78. The runner-up was the Pacific Northwest Chapter team; George E. Mead, 87-6-81; H. K. Mead, 101-20-81; and H. T. Griffith, 111-20-91. A special guest prize was awarded to E. J. Balaam with a low gross score of 81.

Many of the Society members attended the annual dinner and dance of the *HPACCNA* at the St. Francis Hotel at 7:00 p.m. on Monday evening and enjoyed an excellent meal, a variety program of entertainment and dancing in the ballroom.

The Society's Semi-Annual banquet and dance was held in the Palm Court of the Palace Hotel on Wednesday evening at 7:30 p.m., with an excellent dinner, entertainment and dancing. Dr. B. M. Woods, toastmaster, presented President W. L. Fleisher, who introduced the distinguished guests at the speakers' table. M. F. Blankin presented the Research Cup in the absence of the chairman of the Committee on Research, A. E. Stacey, Jr. Mr. Blankin also presented the Eichberg Memorial Cup to the Golden Gate Chapter.

Prof. E. O. Eastwood, First Vice-President of the Society, presented the Past President's Memory Book on behalf of the Council to Dr. F. E. Giesecke, who responded with a brief acknowledgment of his appreciation.

Dr. Woods then presented the speaker of the evening, Dr. C. D. Leake, professor of pharmacology at the University of California Medical School, who gave an interesting address on Human Engineering. Dr. Woods introduced the secretary of the Society and read a resolution presented by the Nominating Committee calling attention to the fact that A. V. Hutchinson has completed 15 years service to the Society as Secretary.

EXPOSITION

When the master switch was pulled on June 20 the great mechanical exhibits at the Pacific Heating and Air Conditioning Exposition ended their short but vibrant life. For five days one of the most important and comprehensive expositions ever held in San Francisco was visited by many thousands of visitors who came to wander, stayed to wonder. And in San Francisco where Nature is generous with its supply of natural heat and air conditioning, Californians and others from the 11 Western states were brought to a dazzling realization of what science and research has accomplished for human need and comfort.

Sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the exposition was acclaimed as one of the greatest contributions to public enlightenment of industrial foresight and progress. According to Charles F. Roth, exposition manager, displays of heating and air conditioning units and equipment became a magnetic attraction to many visitors planning the construction of homes, as well as leading engineers and operators. The exposition proved an added attraction to members and guests of the ASHVE and the *Heating, Piping and Air Conditioning Contractors National Association* whose meetings were held in San Francisco during the same period.

Among the 7,500 who registered were representatives of outstanding construction companies, steamship operators, contractors, engineers, industrial and commercial building owners and operators, home owners, architects, designers, dealers and distributors.

With defense activities approaching rip-tide velocity in California, local and out-of-state contractors evinced intense interest in the exposition. Representatives of many contractor and construction groups attended frequently to survey equipment on display. This interest was accepted as a general reflection of proposals to include modern and efficient heating and air conditioning equipment in all public and private construction projects planned for the immediate future.

Well attended by selected audiences drawn from every part of the United States, the exposition was declared a distinct success, both from the point of view of the exhibitor and appreciative visitors.

The exposition was arranged and managed by Charles F. Roth, President of the International Exposition Co., Grand Central Palace, New York.

SEMI-ANNUAL MEETING 1941

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

PALACE HOTEL, SAN FRANCISCO, CALIF.

June 16-20, 1941

(All Events on Pacific Standard Time)

Sunday, June 15

10:00 A.M. REGISTRATION—Palace Hotel (*Concert Room—Lobby Floor*)

2:00 P.M. Committee Meetings—(See Bulletin Board at Registration Desk)

Monday, June 16

8:30 A.M. REGISTRATION—Palace Hotel (*Concert Room—Lobby Floor*)

10:00 A.M. JOINT SESSION with *Heating, Piping and Air Conditioning Contractors National Association*—St. Francis Hotel (*Colonial Room*) W. L. Fleisher presiding

(*a*) Invocation, Major Harry A. Shuder, U. S. Army Chaplain
(*b*) Singing "America"
(*c*) Greetings from San Francisco
(*d*) Response by J. E. McNevin, President, *HPACCNA*
(*e*) Greetings by Clyde E. Bentley, President, Golden Gate Chapter, and Daniel Hayes, President, *Heating and Piping Contractors Association of San Francisco*
(*f*) Program Announcements
(*g*) Group photograph
(h) Address: A. Appleton, Pres. San Francisco Chapter *AIA*

12:15 P.M. Joint Luncheon—Hotel St. Francis—Tickets $1.25 per person—sponsored by *HPACCNA*

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2:00 P.M. Opening of Pacific Heating and Air Conditioning Exposition—Civic Auditorium

2:00 P.M. Golf Tournament, Lakeside Country Club

2:30 P.M. Council Meeting

7:00 P.M. Dinner and Dance—Hotel St. Francis (Mural Room—Tickets $3.50 per person—sponsored by *HPACCNA*) (ASHVE Members Invited)

Tuesday, June 17

10:00 A.M. Nominating Committee Meeting

10:30 A.M. Ladies' sightseeing trip via Golden Gate Bridge to Marin County, Muir Woods (Redwoods) and Mt. Tamalpais (Luncheon $1.00 at Mt. Tamalpais Tavern—Elevation 1500 ft—Transportation free) Buses will return to Palace Hotel at 4:00 p.m.

1:30 P.M. TECHNICAL SESSION—(*Rose Room—Lobby Floor*)
Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson
Weather Data for Cooling Design Application, by J. D. Kroeker
Concerning Conservation of Underground Water with Suggestions for Control, by Noel E. Porter
Recent Developments in Absorption Refrigeration, by Glen Miller
Amendments to By-Laws

12:00 P.M. Pacific Heating and Air Conditioning Exposition—Civic Auditorium

7:00 P.M. Sightseeing (as selected by individual members; no planned Entertainment)

Wednesday, June 18

9:30 A.M. TECHNICAL SESSION—(*Rose Room—Lobby Floor*)
The Application of Gas Fuel to Existing Heating Boilers, by R. L. Grutzmacher
The Thermal Conductivity of Wood, by J. D. MacLean
Effect of Insulation on Plant Performance in the Research Residence, by A. P. Kratz and S. Konzo
An Analysis of Factors Influencing Building Heat Losses, by Paul D. Close

10:00 A.M. Ladies assemble for scenic ride to the interesting places of San Francisco —last Bus will leave Palace Hotel at 10:30 a.m. and return at 3:00 p.m. (Luncheon $1.00 in *Sequoia Room* of famous Cliff House overlooking Pacific Ocean and Seal Rocks—Transportation free)

12:00 P.M. Pacific Heating and Air Conditioning Exposition—Civic Auditorium

7:00 P.M. Semi-Annual Banquet and Dance—Palace Hotel (*Palm Court*) Toastmaster, Dr. B. M. Woods, University of California—Presentation of Past President's Memory Book to Dr. F. E. Giesecke; "Human Engineering" by Dr. C. D. Leake; (*HPACCNA* members invited)—Tickets at Registration Desk $3.50 per person

Thursday, June 19

9:30 A.M. TECHNICAL SESSION—(*Rose Room—Lobby Floor*)
Resumé of Society Research in Cooperating Institutions, by A. E. Stacey, Jr.
The Interaction Constant for Moist Air, by John A. Goff and A. C. Bates

Local Cooling of Workers in Hot Industry, by F. C. Houghten, M. B. Ferderber and Carl Gutberlet
Economical Air Velocities for Mechanical Air Filtration, by F. B. Rowley and R. C. Jordan

11:30 A.M. Adjournment

11:30 A.M. University of California luncheon-inspection trip via San Francisco-Oakland Bay Bridge—Buses to Claremont Hotel, Berkeley, assemble at New Montgomery entrance Palace Hotel

12:00 P.M. Pacific Heating and Air Conditioning Exposition—Civic Auditorium

12:15 P.M. Luncheon at Claremont Hotel (*Spanish Room*—Transportation and Luncheon ticket $1.75 per person; luncheon ticket only 85¢)

1:30 P.M. Buses leave Claremont Hotel for University Campus, entrance via Sather Gate

2:00 P.M. Ladies' sightseeing trip through University of California Campus and Buildings

2:00 P.M. TECHNICAL SESSION—University of California, Engineering Building (*Room 104*)
Performance Characteristics of a Forced Draft, Counterflow Spray Cooling Tower, by Howard H. Niederman, E. D. Howe, John P. Longwell, Ralph A. Seban and L. M. K. Boelter

3:00 P.M. Inspection trip for men through cyclotron and engineering laboratories

4:00 P.M. Buses will leave Campus for return trip to San Francisco as soon as loaded (last Bus to leave International House after Tea)

4:30 P.M. Tea for members and ladies at International House, University of California

Friday, June 20

ASHVE—JOINT PROGRAM—*HPACCNA*

10:00 A.M. Sightseeing and Inspection Trips

10:00 A.M. Ladies shopping tour in San Francisco downtown district—See famous "Gumps Store" and novelty shops in Chinatown (all within walking distance from Palace Hotel)

10:00 A.M. Golf Tournament (ASHVE *vs.* *HPACCNA*—Leave Palace Hotel by Taxi for Lakeside Country Club (cost $1.00, maximum for 4 persons —Greens fees $2.00—Assemble at Lakeside Locker Room)

12:00 P.M. Pacific Heating and Air Conditioning Exposition—Civic Auditorium

2:00 P.M. Inspection trip for men and women to Pacific Gas and Electric Co., Station "A"—Transportation by Buses

COMMITTEE ON ARRANGEMENTS

G. M. SIMONSON, *General Chairman*

C. E. BENTLEY, *Vice-Chairman* B. M. WOODS, *Vice-Chairman*

Banquet: N. H. Peterson, *Chairman;* Lincoln Bouillon, H. S. Haley, T. A. Marshall, F. F. Urban, W. O. Stewart.

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Entertainment: C. L. Peterson, *Chairman;* W. E. Beggs, A. J. Hess, F. W. Hook, B. W. Moore, C. E. Wayland, C. D. Williams.

Finance: A. J. Bouey, *Chairman;* R. E. Chase, L. H. Cochran, E. E. Goehler, J. I. Krueger, O. W. Ott.

Ladies: J. F. Kooistra, *Chairman;* Mrs. Mason Emanuels, Mason Emanuels, M. J. Hauan, Leo Hungerford, Mrs. J. F. Kooistra, J. P. McDermott, Mrs. T. J. White, T. J. White.

Publicity: E. W. Simons; *Chairman;* C. W. Brissenden, H. T. Griffith, O. W. Johnson, J. F. Park, E. J. Rosen.

Reception: J. A. Hill, *Chairman;* G. K. Brokaw, G. J. Cummings, H. H. Douglas, B. W. Farnes, E. H. Goins, G. H. Harrison, R. B. Holland, S. D. Peterson.

Sports: F. W. Kolb, *Chairman;* James Gayner, Maron Kennedy, E. A. Murhard, G. E. Mead, W. P. Scott, Jr.

Transportation: R. A. Folsom, *Chairman;* W. W. Gee, Jr., R. A. Parker.

Committee Alternates: E. C. Cooley, W. W. Gee, Jr., H. E. Newman, E. J. Ploskey.

No. 1182

PANEL HEATING AND COOLING ANALYSIS

By B. F. Raber * and F. W. Hutchinson,** Berkeley, Calif.

RADIANT heating or cooling for comfort differs from convective heating or cooling primarily in that it attempts to control body heat loss through adjustment of the inside surface temperature rather than by adjustment of the room air temperature. Since only a fraction of the interior surface can be heated or cooled, the temperature of the controlled surface is a function of, but is not equal to, the weighted average or mean radiant temperature (MRT) of the room. Likewise the room air temperature reaches an equilibrium value which is in part determined by the panel temperature.

For the reasons given, the calculations involved in the exact design of a new radiant system or in estimating the characteristics of an existing radiant system are more complex than those needed in convective design. For convective systems an inside design air temperature is selected and a straightforward calculation determines the necessary rate of heat input or removal as a function of the outside temperature; the calculation is simple and direct, also exact within the limits of accuracy of the physical constants.

In panel heating, such a simple, direct and comparably exact solution does not seem feasible. The conditions necessary for comfort dictate a particular relationship between room air temperature (t_a) and MRT, yet neither t_a nor MRT is subject to direct control. For a particular system with a given ventilation rate and panel area, the panel surface temperature is the only variable subject to such control and it must vary as a function of the outside temperature. The basic design problem for such a system is, then, to select a panel surface temperature (t_p) such that the two terms t_a and MRT, (which vary with t_p), will reach such equilibrium values as will retain, with respect to one another, a relationship that will satisfy the criteria for comfort.

The function $t_a = \phi$ (MRT) which correlates conditions of substantially equal comfort has been established experimentally over a limited operating range. It is based essentially on physiological considerations, but as better data become available on the way in which the human body utilizes the three mechanisms of heat dissipation, evaporation, convection and radiation, a rational

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** Assistant Professor, Dept. of Mechanical Engrg., University of California. Member of ASHVE.

Presented at Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, San Francisco, Calif., June, 1941.

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approach based on a heat balance at the skin may become possible. Lacking a more adequate explanation, $t_a = \phi$ (MRT) may be regarded as an empirical special condition arbitrarily imposed on conditions in the room.

But t_a and MRT must have specific values such that a heat balance will be established on the ventilation air and at any part of the interior surface. These equilibrium temperatures can be directly calculated from the heat transfer equations and are obviously dependent on the character of the surfaces and the conductances of the walls, floors, and other units of the construction.

CALCULATION PROCEDURE

Thus for a given room with specified ventilation rate and panel area and a given outside temperature, the necessary panel surface temperature for comfort can be determined, by trial and error, as follows:

1. Assume a panel surface temperature and calculate the corresponding values of t_a and MRT.
2. From the comfort equation $t_a = \phi$ (MRT), using the value of t_a calculated in step 1, determine the value of MRT necessary for comfort.
3. If the values of MRT as calculated in steps 1 and 2 are the same, comfort will be realized at the assumed panel temperature. If the calculated MRT is below that for comfort the room will be too cold; if above, too hot; in this event assume a second value of t_p and repeat the analysis.

From the outline given, a particularly important fact becomes evident: the air temperature corresponding to comfort conditions in a radiant conditioned room will have a different equilibrium value for each value of the outside temperature. Thus, exact control of comfort in such a room cannot be realized by maintaining a constant inside temperature. As a corollary, simplified design methods which propose calculation of the necessary MRT in terms of a specified room air temperature are inherently incorrect since t_a is a variable of the system which can be fixed only in terms of the same variables—outside and panel temperatures—which are involved in the determination of MRT.

Therefore, the criterion of panel design is the condition that the environment be such as to produce equilibrium values of t_a and MRT which will fall on the curve $t_a = \phi$ (MRT), but it is not possible to specify the point on the curve at which the necessary condition will be realized. Fig. 1 shows the two conditions which must be satisfied. As the panel surface temperature is varied the slope of the heat transfer lines varies, but for any given value of t_p there is one and only one value of t_a for which both the heat transfer and comfort relationships are satisfied.

In setting up the equations for the rational design of a panel system the most difficult problem is to determine the mean radiant temperature. If the entire interior surface of the room were a panel and could be controlled at a constant and uniform temperature, that temperature would be equal to the MRT. The average panel in an actual room occupies only a small percentage of the total interior surface. Unheated parts of the surface reach different equilibrium values depending on the rates of convective and conductive loss

or gain, and the position of the surface with respect to the panel and to other unheated surfaces. Since the MRT represents a uniform temperature at which the average radiant flux would be equivalent to that in the actual room, it follows that determination of the MRT requires a knowledge of surface temperatures at all points in the room. The basic engineering problem therefore simplifies to one of determining the surface temperature at any given point in the room.

The four sections of this paper deal with the classification and development of the equations and methods needed in the rational solution of panel design problems. The principal emphasis of this paper is on method of exact treat-

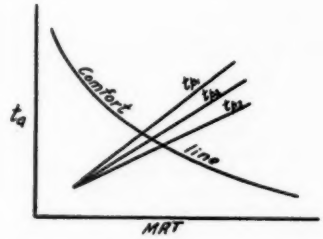

FIG. 1. EQUILIBRIUM VALUES OF AIR TEMPERATURE AND MRT AT INTERSECTIONS WITH PANEL TEMPERATURES

ment. Such a theoretical analysis is justified for at least three eminently practical reasons:

1. Progress in the accurate design of radiant systems depends upon the development of more exact *short cut* methods which will more closely approximate the true condition of the actual system. While the exact analysis is in many instances too burdensome to be of direct application it may nonetheless effectively serve to indicate the direction in which development of approximate methods should proceed.

2. The theoretical analysis is of great practical utility in that it offers a basis of comparison with other methods and thereby directs attention to the effects of such assumptions as are customarily made in practice; it provides a reference point from which the accuracy of simplified solutions can be estimated. Every problem in the engineering research of radiant systems can be located, visualized and evaluated in the basic rational equations. In general, each panel research project has as its aim the simplification or evaluation of a single term or factor in the general theoretical equation.

3. The theoretical equations provide a relatively simple method of evaluating the boundary conditions when estimating the error likely to result from use of an inexact method.

The following development of methods for rational analysis of panel problems is divided into four sections starting with expressions for radiation from a single surface and proceeding to the general case of a heat balance on a

radiant conditioned room with floor, ceiling, partitions, windows and exterior wall surfaces all subject to different heat balances. These sections are:

I. *Energy Leaving a Surface by Radiation:* This section brings together the radiation equations used in calculating the gross and net energy leaving a surface for various special cases and for the general case. The simpler equations are merely defined, but derivations are given for the less common reflected energy equations.

II. *Heat Balance on an Element of an Area:* In section II the equations of the preceding section are applied to the basic problem of determining the surface temperature at any point in various types of ideal systems. Numerical examples are included to show the error which can occur if the equations for a simple case are erroneously applied to an actual room.

III. *Average Surface Temperature as a Function of Average Shape Factor:* This section investigates the adequacy of the usual assumption that the average equilibrium temperature of a surface can be directly calculated from a knowledge of the average shape factor of that surface with respect to the surface from which energy is being received.

IV. *Determination of Equilibrium Conditions in a Radiant Conditioned Room:* The exchange equations of section II are here applied to the problem of calculating the average temperature of each different type of room surface and the inside air temperature. From the calculated surface temperatures the MRT can be evaluated and comfort conditions then checked from a curve similar to that of Fig. 1.

I. *Energy Leaving a Surface, A_1, by Radiation*

In the cases which follow the temperature at a point on the surface A_1 is taken as constant with respect to time, but variable with respect to position. The emissivity, e_1, is taken as constant with respect to both time and position and is considered independent of both wave length and angle of emission. All surfaces considered in the analysis are perfect diffusers.

A) Total energy emitted,
By infinitesimal area dA_1

$$dq = \sigma e_1 T_1^4 dA_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

By area A_1

$$q = \sigma e_1 \int_{A_1} T_1^4 dA_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2$$

where q = Btu per hour.
A = square feet.
σ = Stefan-Boltzman constant = 0.172×10^{-8}.
T = degrees Fahrenheit absolute.

B) Energy emitted by the infinitesimal area dA_1 through the small solid angle $d\omega_1$ and in a direction making an angle ϕ_1 with the normal to dA_1.

$$d^2q = \sigma e_1 T_1^4 dA_1 \frac{\cos \phi_1 d\omega_1}{\pi} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

since $\cos \phi_1 dA_1$ is the projected area of dA_1 normal to the direction of radiation and $\frac{d\omega_1}{\pi}$ is the fraction of hemispherical solid angle through which radiation occurs.

C) Energy leaving A_1 and striking A_2 when $e_2 = 1$.

1. When all other objects which A_1 *sees* are perfect absorbers ($e = 1$).

a) The energy emitted by dA_1 and striking dA_2,

$$d^2q = \sigma e_1 T_1^4 \frac{\cos\phi_1 \cos\phi_2 dA_1 dA_2}{\pi r^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

from Equation (3) noting that,

$$d\omega_1 = \frac{\cos\phi_1 dA_2}{r^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

b) The energy emitted by dA_1 and striking A_2,

$$dq = \sigma e_1 T_1^4 dA_1 \int_{A_2} \frac{\cos\phi_1 \cos\phi_2 dA_2}{\pi r^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where $\cos\phi_1$, $\cos\phi_2$ and r are variables with respect to the position of dA_1 on A_1. (For parallel surfaces—as in Fig. 4 — $\phi_1 = \phi_2$).

c) The energy emitted by A_1 and striking A_2 is then,

$$q = \sigma e_1 \int_{A_1} \int_{A_2} \frac{T_1^4 \cos\phi_1 \cos\phi_2 dA_1 dA_2}{\pi r^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

2. When A_1 is completely enclosed by surfaces all of which, except A_2 are perfect reflectors ($e = 0$).

a) Energy emitted by A_1 and striking A_2 as a result of direct passage from A_1 without reflection; given by Equations (4), (5), or (7).

b) Energy emitted by A_1 and striking A_2 after one reflection from a wall,

1. Energy emitted by A_1 and striking dA_w (from Equation (6),

$$dq = \sigma e_1 dA_w \int_{A_1} \frac{T_1^4 \cos\psi_1 \cos\psi_w dA_1}{\pi r^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

2. Energy reflected by dA_w and striking dA_2,

$$d^2q = \sigma e_1 dA_w \frac{\cos\phi_w \cos\phi_2}{\pi s^2} \int_{A_1} \frac{T_1^4 \cos\psi_1 \cos\psi_w dA_1}{\pi r^2} dA_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

3. Energy reflected by A_w and striking A_2 (= energy emitted by A_1 and striking A_2 after one reflection),

$$q = \sigma e_1 \int_{A_2} \int_{A_w} \left(\frac{\cos\phi_w \cos\phi_2}{\pi s^2} \right) \left(\int_{A_1} \frac{T_1^4 \cos\psi_1 \cos\psi_w dA_1}{\pi r^2} \right) dA_w dA_2 \quad . \quad . \quad . \quad (10)$$

c) Total energy striking A_2 from A_1, including an infinite number of wall reflections: Can be expressed as an integral equation or developed by a continuation of the method used in part "*b*". As will be shown in a later section, the evaluation of terms beyond the first reflection is in many practical cases unnecessary.

D) The total energy leaving A_1 and striking A_2 (including reflections from A_2 of energy initially emitted by A_1, but not including energy reflected by A_1 which is emitted by any other source).

1. When all other objects which A_1 *sees* have $e = 1$.

a) When $e_1 = 1$: Use Equation (7).

b) When $0 < e_1 < 1$,

$$= \frac{\sigma e_1}{\pi} \int_A (a + kb + k^2 c + k^3 d + \ldots\ldots) dA_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

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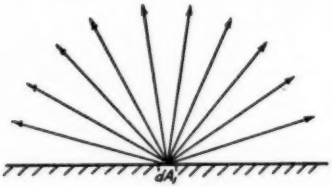

FIG. 2. FOR CONDITIONS UNDER I (A)

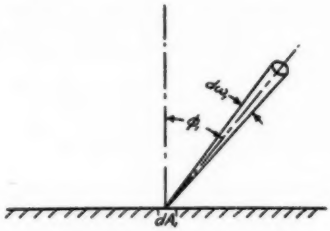

FIG. 3. FOR CONDITIONS UNDER I (B)

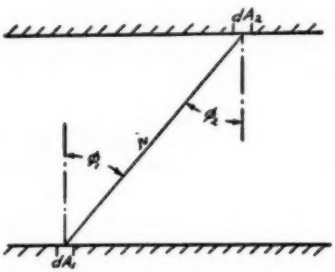

FIG. 4. FOR CONDITIONS UNDER I (C)

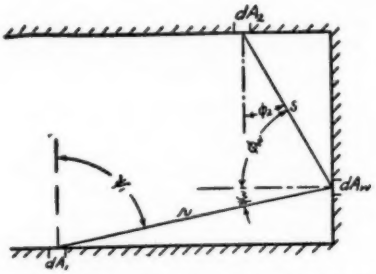

FIG. 5. FOR CONDITIONS UNDER I (C) 2B

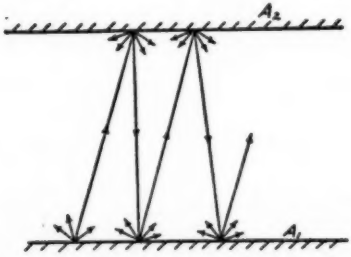

FIG. 6. FOR CONDITIONS UNDER I (D)

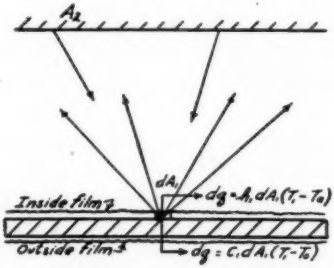

FIG. 7. FOR CONDITIONS UNDER II (A) 2

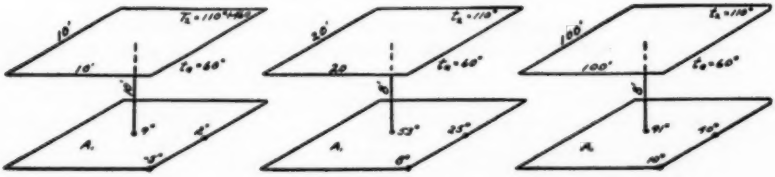

FIG. 8, a, b, c. FOR CONDITIONS UNDER II (A) 3

(See derivation in Appendix A) *where,*

$$k = \frac{(1 - e_1)(1 - e_2)}{\pi^2}$$

$$a = \int_{A_1} \frac{T_1^4 \cos \phi_1 \cos \phi_2 dA_1}{r^2}$$

$$b = \int_{A_1} \left[\frac{\cos \phi_1 \cos \phi_2}{r^2}\right] \left[\int_{A_2} \left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right) (\text{Term } "a") \, dA_2\right] dA_1$$

2. When all other objects which A_1 *sees* are perfect reflectors ($e = 0$).

a) Energy striking A_2 as a result of direct passage (including all direct reflections) from A_1 to A_2: Use Equation (11).

b) Energy striking A_2 after one reflection from a wall and without limit to the number of reflections between A_1 and A_2 directly. Combining Equations (10) and (11) after revising (11) to get the energy leaving dA_1,

$$q = \frac{\sigma e_1}{\pi} \int_{A_2} \int_{A_w} \left[\frac{\cos \phi_w \cos \phi_2}{\pi s^2}\right] \left[\int_{A_1} \left(\frac{\cos \psi_1 \cos \psi_w}{\pi r^2}\right) (T_1^4 + ka' + k^2 b' + \ldots.) dA_1\right] dA_w dA_2 \quad . \quad . \quad (12)$$

$$a' = \int_{A_1} \left[\frac{\cos \phi_1 \cos \phi_2}{r^2}\right] \left[\int_{A_1} \frac{T_1^4 \cos \phi_1 \cos \phi_2 dA_1}{r^2}\right] dA_2$$

$$b' = \int_{A_2} \left[\frac{\cos \phi_1 \cos \phi_2}{r^2}\right] \left[\int_{A_1} \left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right) (\text{Term } a') \, dA_1\right] dA_2$$

c) Energy striking A_2 after multiple wall reflections: Energy will be returned to A_1 from A_2 by way of wall reflections; the quantity so returned must be added to the right side of Equation (11) to get the total energy from A_1 and this total can then be used in an expression of the same form as Equation (12) but carried out to n reflections. The equation for this case can be set up in the form of an integral equation, but its solution is so complex that the expression is of little utility.

II. *Heat Balance on an Element of Area, dA_1*

The element of area dA_1 is exchanging energy by radiation with a surface A_2 which is at a constant and uniform temperature T_2, by convection with room air at temperature T_a and by conduction and convection through the slab with outside air at temperature T_o; all temperatures are expressed in degrees Fahrenheit absolute. Conduction along the slab is taken as zero.

A) When $e_1 = e_2 = 1$ and dA_1 *sees* only A_2 and absolute space (equivalent to *seeing* a perfectly absorbing surface at a temperature of absolute zero).

1. The net radiant gain of energy at dA_1 is equal to the energy absorbed at dA_1 from A_2 less the total energy emitted by dA_1, and from Equations (1) and (4),

$$= \sigma T_2^4 dA_1 \left(\int_{A_2} \frac{\cos \phi_1 \cos \phi_2 dA_2}{\pi r^2}\right) - \sigma T_1^4 \, dA_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

The term under the integral is a function only of the geometry of the system and, for convenience, the following definition is used,

$$F_{A_2, dA_1} = \int_{A_2} \frac{\cos \phi_1 \cos \phi_2 dA_2}{\pi r^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

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where F_{A_2, dA_1} is the *shape factor* and is equal to $\frac{A_2}{dA_1}$ multiplied by the fraction of energy leaving A_2 which falls on dA_1. Equation (13) then becomes,

$$= \sigma dA_1 (T_2^4 F_{A_2, dA_1} - T_1^4) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

where the shape factor can be evaluated for any particular system.[1]

2. Heat balance on dA_1:

$$\sigma dA_1(T_2^4 F_{A_2, dA_1} - T_1^4) = h_1 dA_1(T_1 - T_a) + C_1 dA_1(T_1 - T_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

where

h_1 = film coefficient of heat transfer from surface dA_1 to room air.
C_1 = conductance of slab and outside air film from surface dA_1 to outside air.

3. Numerical example. Consider a slab of area A_1 uniformly heated to a surface temperature of 110 F and located at distance L opposite and parallel to an unheated slab of equal area. The unheated surface is perfectly insulated, conduction term drops out of Equation (16), and transfers heat by convection with room air at 60 F. The film transfer coefficient is taken as 0.75 (the commonly used film coefficient of 1.65 includes

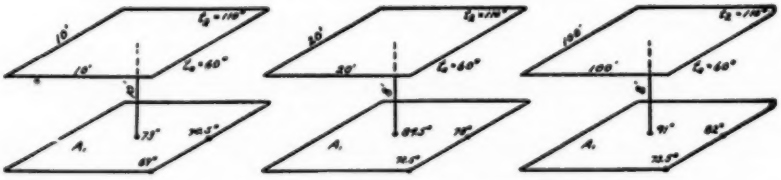

FIG. 9, a, b, c. FOR CONDITIONS UNDER II (B) 2

radiant as well as convective heat transfer; it should not be used in radiant heating problems).

Equation (16) can be used to determine the equilibrium temperature at various points on the unheated surface. Figs. 8a, b, c, show equilibrium temperatures at the center, middle of each side and corner, of the unheated slab for different arrangements. Note that the temperatures shown are determined by the geometry of the arrangement rather than by the actual dimensions of the surface. Thus in Fig. 8a the temperatures would remain the same if the surfaces were 20 ft by 20 ft or 100 ft by 100 ft, providing the distance between them is changed to 20 ft and to 100 ft, respectively.

In practice, a system in which A_1 would see absolute space could never be realized and equilibrium temperatures as low as those of Fig. 8 would not occur. However, if absolute space is replaced by a clear night sky, the nocturnal radiation from A_1 can be measured or can be roughly calculated by assuming an equivalent temperature of space as $-$ 50 F[2] and for such an actual system the equilibrium temperatures will remain lower than the temperature of the air in contact with the surface.

B) When $e_1 = e_2 = 1$ and dA_1 is enclosed by surfaces all of which, except A_2, are at the same temperature as dA_1 and have $e = 1$. (These conditions define a system in which the only net radiant transfer to or from dA_1 is with respect to direct exchange with A_2.)

[1] See *Heat Transmission* by McAdams.
[2] See *The Calculation of Heat Transmission*, by Fishenden and Saunders.

1. Heat balance. From Equations (6), (14), and (16) noting that,

$F_{a_2 da_1} dA_1 = F_{da_1 a_2} A_2$ obtain . . . . . (17)

$\sigma F_{a_2 da_1} dA_1 (T_2^4 - T_1^4) = h_1 dA_1 (T_1 - T_a) + C_1 dA_1 (T_1 - T_o)$ . . . . . (18)

2. Numerical example. Using Equation (18) with the surface arrangements of Fig. 8, results in Fig. 9 a, b, c.

C) When $e_1 = e_2 = 1$ and dA_1 is enclosed by surfaces all of which, except A_2, are perfect reflectors. (These conditions define a system in which all energy emitted by dA_1 is absorbed by A_2 or is later reabsorbed by A_1 after reflection.)

Net transfer from A_2 to dA_1
= direct transfer without reflection (a in Fig. 10)
+ transfer as result of one reflection (b in Fig. 10)
+ transfer as result of two reflections (c in Fig. 10)
+ transfer as result of three reflections
+ transfer as result of n reflections

1. Direct transfer a can be evaluated from Equation (6) [it is equal to the radiation term in Equation (18)].

2. Transfer as a result of one reflection, b, is from Equation (10),

$$dq = \sigma(T_2^4 - T_1^4)\left[\int_{A_w}\left(\frac{\cos\psi_1 \cos\psi_w}{\pi r^2}\right)\left(\int_{A_1}\frac{\cos\phi_2 \cos\phi_w \, dA_2}{\pi s^2}\right)dA_w\right]dA_1 \quad . \quad . \quad (19)$$

$$= \sigma(T_2^4 - T_1^4)\, dA_1 \int_{A_w}(F_{a_2 da_w})\left(\frac{\cos\psi_1 \cos\psi_w}{\pi r^2}\right)dA_w \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

For a small but finite area of the wall, ΔA_w, the shape factor F_{a2da_w} remains substantially constant and can be taken out from under the integral sign to give,

$$dq = \sigma(T_2^4 - T_1^4)\, dA_1 \sum_{\Delta A_w}^{A_w}(F_{a_2 \Delta a_w})\left(\int_{A_w}\frac{\cos\psi_1 \cos\psi_w \, dA_w}{\pi r^2}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

or

$$dq = \sigma\,(T_2^4 - T_1^4)\, dA_1 \sum_{\Delta A_w}^{A_w}(F_{a_2 \Delta a_w})(F_{\Delta a_w dA_1}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Equation (22) can be more simply written,

$$dq = \sigma(T_2^4 - T_1^4)\, dA_1 \cdot F_{a2} dA_1 r_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

where $F_{a2} dA_1 r_1$ is $\dfrac{A_2}{dA_1}$ multiplied by the fraction of energy emitted by A_2 which strikes dA_1 after one wall reflection and it is defined by the equation,

$$F_{a2} dA_1 r_1 = \sum_{\Delta A_w}^{A_w}(F_{a_1 \Delta A_w})\,(F_{\Delta a_w dA_1}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

3. The transfer of energy resulting from further reflections can be expressed as,

(a) For two reflections, $dq = \sigma(T_2^4 - T_1^4)\, dA_1 \cdot F_{a_2 da_1 r_2}$ . . . . . (25)

(b) For n reflections,

$dq = \sigma\,(T_2^4 - T_1^4)\, dA_1 \cdot F_{a_2 da_1 r_n}$ . . . . . (26)

where the terms $F_{a_2 da_1 r_n}$ can be determined by analysis as in b above.

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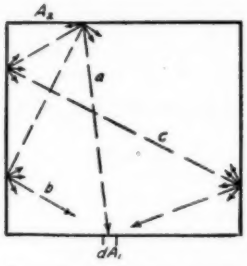

FIG. 10. FOR CONDITIONS UNDER II (C)

4. The total energy transferred from A_2 to dA_1, is then,

$$dq = \sigma\,(T_2^4 - T_1^4)\,dA_1\,(F_{A_2dA_1} + F_{A_2dA_1r_1} + F_{A_2dA_1r_2} + \ldots\ldots\ldots + F_{A_2dA_1r_n}) \quad . \quad (27)$$

5. Numerical example. Fig. 11*a* shows the variation of shape factor for the direct transfer of energy from A_2 to A_1, first term in series of Equation (27); the system is a room 20 ft × 20 ft × 8 ft and is shown in elevation in Fig. 11*c*. Fig. 11*b* shows variation of the shape factor for energy which strikes A_1 after one reflection, second term in the series of Equation (27).

From the values plotted in Figs. 11*a* and 11*b* it is evident that the distribution of energy from A_2 and falling on dA_1 is extremely variable with respect to the position of dA_1 for energy which is transferred directly, but is practically uniform for energy received after one reflection. Since the degree of uniformity must increase as the number of reflections increases it follows that for this particular case a satisfactory and more direct solution can be obtained by considering that all energy except that which is directly transferred is uniformly received over A_1. With this assumption and taking the overall shape factor of A_1 with respect to A_2 as a constant having the calculable value K, Equation (27) becomes,

$$dq = \sigma\,(T_2^4 - T_1^4)\,dA_1\left[F_{A_2dA_1} + \left(K - \frac{\int_{A_1} F_{A_2dA_1}\,dA_1}{A_1}\right)\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

If $F_{A_2dA_1}$ is taken as constant over the small area ΔA_1,

$$\sum^{A_1} F_{A_2\Delta A_1}$$

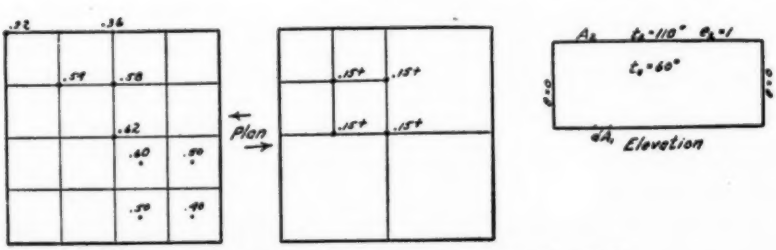

FIG. 11, a, b, c. FOR CONDITIONS UNDER II (C) 5

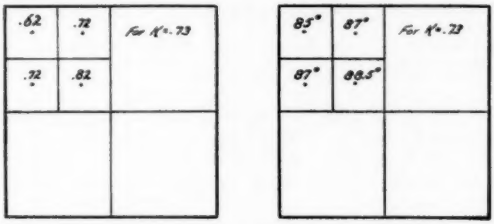

FIG. 12, a, b. FOR CONDITIONS UNDER II (C) 5

$$dq = \sigma\,(T_2^4 - T_1^4)\,dA_1\left[F_{A_2dA_1} + \left(K - \frac{\Delta A_1}{A_1}\right)\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (29)$$

Evaluating terms of Equation (29) from the shape factors given in Fig. 11,

$$\frac{\sum_{\Delta A_1}^{A_1} F_{A_2\Delta A_1}}{A_1} = \frac{4\,(0.62 + 0.50 + 0.50 + 0.40)\,25}{20 \times 20} = 0.51, \text{ so,}$$

$dq = \sigma\,(T_2^4 - T_1^4)\,dA_1\,[F_{A_2\Delta A_1} + (K - 0.51)]$
which can be written,

$$dq = \sigma\,(T_2^4 - T_1^4)\,dA_1 F_{A_2\Delta A_1 0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (30)$$

where $F_{A_2\Delta A_1 0}$ is an equivalent shape factor, including reflections, for any particular element of area ΔA_1. Numerical values of $F_{A_2\Delta A_1 0}$ at various points on A_1 are shown in Fig. 12*a* while the corresponding equilibrium temperatures, calculated from the heat balance equation,

$$\sigma\,(T_2^4 - T_1^4)\,dA_1 F_{A_2\Delta A_1 0} = h_1 dA_1\,(T_1 - T_a) + C_1 dA_1 (T_1 - T_o) \quad . \quad . \quad . \quad . \quad . \quad (31)$$

are shown in Fig. 12*b*. Of particular interest is the fact that the equivalent shape factor at the center of A_1 (Fig. 12*a*) would exceed unity if the walls were perfect specular reflectors (instead of reradiating matt surfaces from which the distribution of reflected energy is independent of angle).

D) When $e_1 = e_2 = 1$ and dA_1 is enclosed by surfaces all of which, except A_2, have e_w between zero and unity. The walls are considered to be at the same temperature as the particular element of area, dA_1, on which the heat balance is being established.

Using the method of analysis developed for perfectly reflecting walls, but noting that the actual wall reflectivity, $(1 - e_w)$, becomes a multiplying factor in Equation (9), it is found that Equation (27) becomes:

$$dq = \sigma(T_2^4 - T_1^4)dA_1\,[F_{A_2dA_1} + (1 - e_w)F_{A_2dA_1r_1} + \ldots + (1 - e_w)^n F_{A_2dA_1r_n}] \quad . \quad . \quad (32)$$

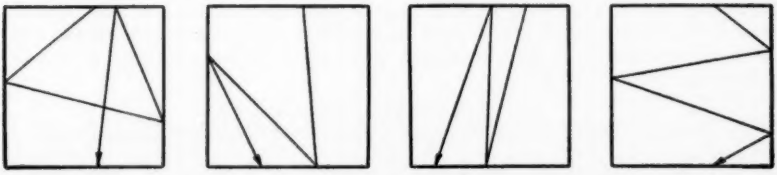

FIG. 13, a, b, c, d. FOR CONDITIONS UNDER II (F)

and the heat balance on dA_1 is therefore given by,

$$\sigma(T_2^4 - T_1^4)dA_1 [F_{\text{ad}A_1} + (1 - e_w) F_{\text{ad}A_1r_1} + \ldots + (1 - e_w)^n F_{\text{ad}A_1r_n}]$$
$$= h_1 dA_1(T_1 - T_a) + C_1 dA_1(T_1 - T_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (33)$$

Equation (33) is not subject to treatment by the simplification of Equation (28) and its solution depends on evaluation of the series.

E) When e_1 and e_2 are each between zero and unity and A_1 is enclosed by surfaces all of which, except A_2, have $e_w = 1$ and are at a temperature equal to that of dA_1.

The net radiant heat transfer is evaluated in a manner similar to that used in deriving Equation (11); the resultant heat balance is:

$$\frac{\sigma e_1}{\pi}\left[T_2^4(a + kb + k^2c + \ldots) - T_1^4 \int_{A_2} (a' + kb' + k^2c' + \ldots)dA_2\right]dA_1$$
$$= h_1 dA_1(T_1 - T_a) + C_1 dA_1(T_1 - T_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

$$\text{where, } a = \int_{A_1} \frac{\cos \phi_1 \cos \phi_2 dA_2}{r^2}$$

$$a' = \frac{\cos \phi_1 \cos \phi_2}{r^2}$$

$$b = \int_{A_2}\left[\frac{\cos \phi_1 \cos \phi_2}{r^2}\right]\left[\int_{A_1}\left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right)(\text{Term } a)dA_1\right] dA_2$$

$$b' = \int_{A_1}\left[\frac{\cos \phi_1 \cos \phi_2}{r^2}\right]\left[\int_{A_2}\left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right)(\text{Term } a')dA_2\right] dA_1$$

$$k = \frac{(1 - e_1)(1 - e_2)}{\pi^2}$$

F) When e_1, e_2 and e_w are all between zero and unity (not necessarily equal to one another) and A_1 is enclosed by surfaces all of which, except A_2, are at a temperature equal to that of the element of area dA_1 on which the heat balance is to be established.

Fig. 13 shows some of the paths followed by energy initially emitted by dA_1. Fig. 13*a* shows a typical path for that part of the energy emitted by dA_1 which reaches A_2 after one or more reflections from the wall surfaces; this fraction can be evaluated by a series of equations of the form of Equation (10). Fig. 13*b* shows path for the energy transferred as a result of reflections between A_1 and A_2; this fraction is evaluated in Equation (11). Figs. 13*c* and *d* show possible paths involving reflections between walls and either A_1 or A_2; evaluation of transfer by such paths is possible, but too complex to be of great practical value.

Since the system described more closely approaches the conditions of an actual room than do any of the preceding cases, it follows that an exact analytical solution of practical problems is usually not feasible; the practical problem must be solved by setting up equations for some less complex ideal system. The more closely the ideal system approaches the real system, the less will be the error. The section which follows will show by example the magnitude of the error which can result from over simplification of the actual system.

III. *Average Surface Temperature as a Function of Average Shape Factor*

A procedure commonly employed in practical problems is to calculate an average surface temperature based on an average value of the shape factor of

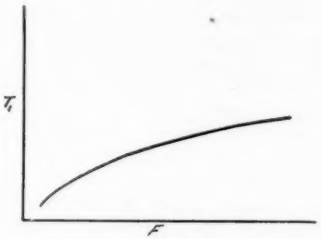

FIG. 14. FOR CONDITIONS UNDER III

that surface with respect to the surface with which it is transferring energy. The adequacy of this method can be investigated by examining any one of the heat balance equations. Consider Equation (18),

$$\sigma F_{A2dA1}\, dA_1\, (T_2^4 - T_1^4) = h_{1d} A_1\, (T_1 - T_a) + C_1 dA_1 - (T_1 - T_o)$$

If all terms in this equation other than T_1 and F_{A2dA1} are known, it becomes possible to calculate the equilibrium value of T_1 corresponding to any particular shape factor. From a series of such calculations data are obtained to plot T_1 as a function of the shape factor. The position and shape of the resultant curve (see Fig. 14) is dependent on the numerical value of the terms of the heat balance equation, but for the present discussion the important fact is that T_1 is shown to be a non-linear function of the shape factor. This being so, the true average of point temperatures on the surface is not equal to the temperature corresponding to the average of point shape factors, or

$$\frac{\int_A F dA}{A} \neq \frac{K \int_A T dA}{A} \qquad (35)$$

In the usual operating range of low temperature panels the curve of Fig. 14 so closely approximates the function, $T_1^4 = \phi(F_{A2dA1})$ that the two curves may be considered as superimposed. Thus the mean effective radiation from A_1 is not a linear function of the shape factor or,

$$\frac{\int_A F dA}{A} \neq \frac{\int_A T^4 dA}{A} \qquad (36)$$

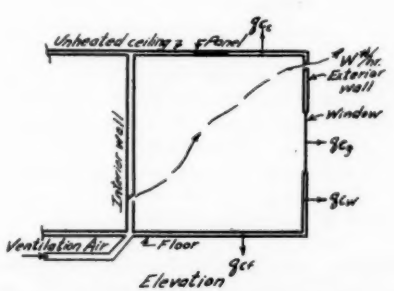

FIG. 15. FOR CONDITIONS UNDER IV (B)

so the temperature determined from the average shape factor is not the true average for the surface nor is it the equivalent temperature at which a uniformly heated surface would emit an amount of energy equal to the emission from the actual infinite number of point sources at different temperatures.

The determination of a curve similar to that of Fig. 14 is easily obtained for any specific set of conditions; its construction will eliminate need for the assumption of an equivalent average temperature, since the temperatures at points can be directly obtained from the curve (when shape factors are known) and the true equivalent temperature then calculated by the equation,

$$T_1 \text{ equivalent} = \frac{\sum_{\Delta A_1}^{A_1} T_1^4 \Delta A_1}{A_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

Further, use of such a curve will show the magnitude of the error likely to result from the approximate method; in many cases the difference is not great enough to justify the added work of a rigorous solution.

IV. *Determination of Equilibrium Conditions in a Radiant Conditioned Room*

A) General statement of the problem.

In any room heated or cooled by radiant means each point on an unheated interior surface will reach an equilibrium temperature corresponding to particular and known values of the panel surface temperature, t_p and the outside air temperature, t_o; these equilibrium temperatures can be evaluated by setting up a heat balance at each surface and on the ventilation air as it passes through the room. Preceding sections have shown that the temperature on an unheated surface is not uniform, but varies with the extent to which each point on the surface sees the panel. Thus in a closed room there are an infinite number of unknown surface temperatures and an exact analysis of surface temperatures in such a room would require the simultaneous solution of an infinite number of equations.

While an exact solution is obviously impractical, a solution accurate to any prescribed limit can be obtained by breaking up the unheated interior surface into small areas ΔA and considering the surface temperature constant over each such area. The smaller ΔA becomes, the more exact will be the solution.

In many cases the most significant variation in surface temperature will occur when the conductance and film coefficient change and when the general position of the surface in question is drastically altered with respect to the panel. Where this is the case a substantially accurate analysis of the surface temperatures throughout the room can be made by taking ΔA as equal to the total area of each different type of surface; the analysis will then require simultaneous solution of as many heat balance equations as there are different surfaces.

The method of solution outlined will determine an average temperature of each inside surface. As has been shown in Section III, the average temperature calculated in this way is not exactly equal to the true average of the

point temperatures across the surface, but the difference is sufficiently small to be neglected in many problems. Knowing the average temperatures of all surfaces (and the air temperature) it becomes possible to calculate point temperatures on any one of these surfaces. Continuing in this way isothermals can be located on all surfaces and values of the previously determined average temperatures corrected for the non-linear relationship between,

$$\frac{\int_A FdA}{A} \text{ and } \frac{\int_A TdA}{A}$$

Using the corrected surface temperature it is possible to calculate the mean radiant temperature. The mean radiant temperature and air temperature together establish the environment of the room; if the intersection of the calculated MRT and t_a temperature falls on the comfort line, when this line is plotted as $\text{MRT} = \phi(t_a)$ the panel will provide comfort. If the MRT and t_a lines do not intersect on the comfort line, a change in the heat transfer

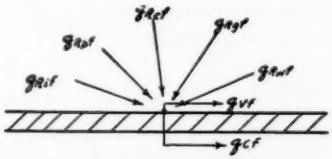

FIG. 16. FOR CONDITIONS UNDER IV (B) 2

characteristics of the system will be necessary; this can be accomplished by changing the ventilation rate, panel area or panel surface temperature.

B) Determination of average surface temperatures.

Consider a room heated or cooled by one or more ceiling panels at constant and uniform surface temperature, t_p. The panels will see the following different types of surfaces: exterior wall, windows, floor, inside partitions; the unheated portion of the ceiling constitutes a sixth type of surface, but one that is not *visible* to the panels. No specification will be made concerning the distribution of surfaces, but it will be considered that (except for panel and unheated ceiling) each surface is able to see five other types of surfaces. Surfaces of the same type will be considered as at the same temperature regardless of the fact that they may be in different walls. In the following treatment each type of surface will be treated as though it occurred only once in the system; for windows, etc., which appear in more than one wall the necessary extension of this treatment is obvious. The equations will be established for a room in which all surfaces have an emissivity of unity (for most surfaces found in practice the emissivity exceeds 0.9).

1. *Nomenclature.*

q = Btu per (hr) (sq ft)

First subscript:

C = conduction
V = convection
R = radiation
W = pounds per hour of ventilation air passing through the room.

Second subscript:

f = floor
w = outside wall
g = glass
c = unheated ceiling
i = interior wall adjoining heated room
p = panel
o = outside air
a = room air

Examples:

q_{cg} = Btu per (square foot) (hour) transfer by conduction between the inside surface of the glass and the outside air.

q_{rpf} = Btu per (square foot) (hour) transfer by radiation from the panel to the floor, based on an area of the floor.

h_f = film coefficient for heat transfer by convection from floor surface to air in room.

C_f = conductance from inside surface of floor to outside air

2. *Heat balance on each type of unheated surface.*

a) Heat balance on the floor.

Net transfer to floor by radiation
= transfer from floor by convection
+ transfer from floor by conduction.
or, from Fig. 16,

$$q_{rif} + q_{rpf} + q_{rcf} + q_{rgf} + q_{rwf} = q_{vf} + q_{cf} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

but by Equation (7),

$$q_{rif} = \sigma(T_i^4 - T_f^4) \int_{Ai} \int_{Af} \frac{\cos \phi_i \cos \phi_f dA_i dA_f}{\pi r^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

$$= \sigma(T_i^4 - T_f^4) F_{if} A_f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (40)$$

$$F_{if} = \frac{1}{A_f} \int_{Ai} \int_{Af} \frac{\cos \phi_i \cos \phi_f dA_i dA_f}{\pi r^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

Since the convection and conduction terms of Equation (38) are expressed in terms of the first power of the temperature, it would simplify the solution if the radiation terms could also be written in terms of the first power. This can be accomplished by defining an equivalent transfer coefficient for radiation by the equation,

$$h_r = \frac{\sigma(T_i^4 - T_f^4)}{T_i - T_f} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (42)$$

$$q_{rif} = \sigma(T_i^4 - T_f^4) F_{if} A_f = h_r (T_i - T_f) F_{if} A_f = h_r (t_i - t_f) F_{if} A_f \quad . \quad . \quad . \quad . \quad . \quad . \quad (43)$$

When approximate values of t_i and t_f are known or can be estimated, an accurate evaluation of h_r is possible—lacking such knowledge of temperatures h_r can be taken as approximately equal to 1.08; for the average temperature range of low temperature panel heating or cooling this value is accurate to within 4 per cent.

Substituting h_r in Equation (40), and taking h_r as substantially constant for the conditions of the room,

$$h_r A_1 \Big[F_{if}(t_i - t_f) + F_{pf}(t_p - t_f) + F_{cf}(t_c - t_f) + F_{gf}(t_g - t_f) + F_{wf}(t_w - t_f) \Big]$$

$$= \Big[h_f(t_f - t_a) + C_f(t_f - t_o) \Big] A_f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (44)$$

$$t_f = \frac{h_n[F_{if}t_i + F_{pf}t_p + F_{cf}t_c + F_{gf}t_g + F_{wf}t_w] + h_f t_a + C_f t_o}{h_n(F_{if} + F_{pf} + F_{cf} + F_{gf} + F_{wf}) + h_f + C_f} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (45)$$

Since the entire field of vision of the floor is taken up with the five surfaces whose shape factors with respect to the floor appear in the denominator of Equation (45), it follows that,

$$F_{if} + F_{pf} + F_{cf} + F_{gf} + F_{wf} = 1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

$$t = \frac{h_n(F_{if}t_i + F_{pf}t_p + F_{cf}t_c + F_{gf}t_g + F_{wf}t_w) + h_f t_a + C_f t_o}{h_n + h_f + C_f} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (47)$$

b) Heat balance on exterior wall surface. By analogy with Equation (45),

$$t_w = \frac{h_n(F_{fw}t_f + F_{iw}t_i + F_{pw}t_p + F_{cw}t_c + F_{gw}t_g) + h_w t_a + C_w t_o}{h_n(F_{fw} + F_{iw} + F_{pw} + F_{cw} + F_{gw}) + h_w + C_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad (48)$$

In this case, unlike that of the floor, the entire field of vision of the exterior wall may not be occupied by the other five surfaces, as there may be sections of exterior wall in more than one wall of the room. If some of this exterior wall is able to see another part of the same type of surface then,

$$F_{fw} + F_{iw} + F_{pw} + F_{cw} + F_{gw} = 1 - F_{ww} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

where F_{ww} is the shape factor of exterior wall with respect to such portions of the same type of surface as may be in other walls of the room. Note that all shape factors for this case should be expressed in terms of the total area of the exterior wall surfaces. Then from Equation (48)

$$t_w = \frac{h_n(F_{fw}t_f + F_{iw}t_i + F_{pw}t_p + F_{cw}t_c + F_{wg}t_g) + h_w t_a + C_w t_o}{h_n(1 - F_{ww}) + h_w + C_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad (50)$$

Note that if there is exterior wall surface in only one wall of the room the term $F_{wg}t_g$ drops out because the exterior wall is then unable to see the window.

c) Heat balance on glass. By analogy from Equation (50),

$$t_g = \frac{h_n(F_{wg}t_w + F_{fg}t_f + F_{ig}t_i + F_{pg}t_p + F_{cg}t_c) + h_g t_a + C_g t_o}{h_n(1 - F_{gg}) + h_w + C_w} \quad . \quad . \quad . \quad . \quad . \quad . \quad (51)$$

Note that if there is only one exterior wall in the room the term $F_{wg}t_w$ drops out because the window cannot then see the exterior wall.

d) Heat balance on unheated area of ceiling. By analogy from Equation (47) and noting that this balance includes one less term than the preceding ones because the unheated ceiling is unable to see the panel,

$$t_c = \frac{h_n(F_{gc}t_g + F_{wc}t_w + F_{fc}t_f + F_{ic}t_i) + h_c + t_a + C_c t_o}{h_n + h_c + C_c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (52)$$

e) Heat balance on inside partition. By analogy from Equation (47),

$$t_i = \frac{h_n(F_{gi}t_g + F_{wi}t_w + F_{fi}t_f + F_{pi}t_p + F_{ci}t_c) + h_i t_a + C_i t_o}{h_n + h_i + Ci} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (53)$$

f) Heat balance on air passing through the room,

$$0.24W(t_a - t_o) = h_p(t_p - t_a)A_p + h_c(t_c - t_a)A_c + h_g(t_g - t_a)A_g + h_w(t_w - t_a)A_w + h_f(t_f - t_a)A_f + h_i(t_i - t_a)A_i \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (54)$$

$$t_a = \frac{h_p t_p A_p + h_c t_c A_c + h_g t_g A_g + h_w t_w A_w + h_f t_f A_f + h_i t_i A_i + 0.24wt_o}{h_p A_p + h_c A_c + h_g A_g + h_w A_w + h_f A_f + h_i A_i + 0.24W} \quad . \quad . \quad . \quad . \quad (55)$$

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C) Determination of mean radiant temperature (MRT).

From section B six independent Equations (47), (50) to (53), and (55) are available which relate the six unknown temperatures, t_f, t_w, t_g, t_c, t_l, and t_a. For any particular room all of the shape factor terms of these equations can be evaluated from the geometry of the arrangement. The convection and conduction terms require a knowledge of the inside convective film coefficient and the conductance from each inside surface to the outside air.

Then for any room, with the outside temperature known and the panel surface temperature fixed, the inside air temperature and the average temperature of each different surface in the room can be determined by a simultaneous solution of the six independent heat balance equations.

The mean radiant temperature is then calculated from the equation,

$$\text{MRT} = \frac{(T_f^4 A_f + T_w^4 A_w + T_g^4 A_g + T_c^4 A_c + T_l^4 A_l + T_p^4 A_p)^{\frac{1}{4}}}{\text{Total area of interior surface}} - 460 \quad . \quad . \quad . \quad (56)$$

In many problems an accurate approximation can be obtained from the equation,

$$\text{MRT} = \frac{t_f A_f + t_w A_w + t_g A_g + t_c A_c + t_l A_l + t_p A_p}{\text{Total area of interior surface}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (57)$$

Conditions in the room have now been established and reference to the comfort curve (Fig. 1) will determine whether or not the values of t_a and MRT which must result in order to satisfy the heat transfer relationships actually satisfy the conditions necessary for comfort (that is, fall on the curve of Fig. 1). If comfort conditions do not exist it is then necessary to vary the panel surface temperature, the panel area or the ventilation rate and repeat the analysis.

In the usual installation the panel area and ventilation rate are fixed so the panel surface temperature is the only variable subject to control. For such a case the equilibrium inside air temperature corresponding to a condition of comfort must vary with the outside temperature; in any particular installation the method of analysis indicated above makes it possible to determine t_a as a function of t_o.

All of the previous discussion has been with reference to the *average* comfort environment in a radiant conditioned room. A factor of importance which has not yet been considered is the variation in uniformity of irradiation and directional irradiation experienced by an occupant as he moves from place to place in the room. A rational approach to this problem is possible by setting up an energy balance at various positions in the room and at different points on the body surface of the occupant.

CONCLUSION

An attempt has been made to outline a rational method of analyzing panel heating problems and to set up general equations leading to a determination of the mean radiant and inside air temperatures corresponding to a fixed panel temperature and a known outside temperature. It has been shown:

1. That the air temperature in a radiant conditioned room is a function of the MRT and cannot be arbitrarily fixed if comfort is to be realized.

2. That the MRT can be calculated by solving simultaneously the heat balance equations for each different type of interior surface and for the ventilation air.

3. That the error likely to result from application of equations for a simple ideal system to a complex real system may be great.

As already stated, the intent of this paper is to outline the basic theory of radiant heating or cooling calculations rather than to develop methods for practical design. From the theoretical equations, the problems of practical design can be identified and their relative importance evaluated within limits. Working from the theoretical equations for the exact and general case it is possible to develop methods of expressing such necessary information as the shape factors or the series representing reflections in an actual room in terms of either simplified equations or graphs that are of direct and simple application to the problems of practical design.

The extent to which rational methods can be utilized in establishing a simple and direct practical design procedure is dependent in part on a better understanding of the physical significance of terms in the general equations but to a greater extent on the amount of experimental data available to check the accuracy and adequacy of such methods as may, from theory alone, seem to offer some promise. The authors are now at work on a comprehensive panel research program being conducted at the University of California at Berkeley as a cooperative project with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Progress reports will be issued at intervals.

APPENDIX

A) Energy leaving A_1 (of that initially emitted by A_1) and striking A_2, when e_1 and e_2 are between zero and unity (but not necessarily equal to one another). Walls perfect absorbers.

1. Energy emitted by A_1 and striking dA_2

$$dq = \frac{\sigma e_1}{\pi} \int_{A_1} \frac{T_1^4 \cos \phi_1 \cos \phi_2 dA_1}{r^2} \cdot dA_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1a)$$

2. Energy reflected by A_2 and striking dA_1 (1st reflection),

$$dq_{r1} = \frac{\sigma e_1}{\pi} \left(\frac{1 - e^2}{\pi}\right) \int_{A_2} \left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right) \left(\int_{A_1} \frac{T_1^4 \cos \phi_1 \cos \phi_2 dA_1}{r^2}\right) dA_2 \cdot dA_1 \quad . \quad (2a)$$

3. Energy reflected (2nd) by A_1 and striking dA_2,

$$dq_{r2} = \frac{\sigma e_1}{\pi} \frac{(1-e_2)}{\pi} \frac{(1-e_1)}{\pi} \int_{A_1} \left[\frac{\cos \phi_1 \cos \phi_2}{r^2}\right] \left[\int_{A_2} \left(\frac{\cos \phi_1 \cos \phi_2}{r^2}\right) \left(\int_{A_1} \frac{T_1^4 \cos \phi_1 \cos \phi_2 dA_1}{r^2}\right) dA_2\right] dA_1 \cdot dA_2 \quad . \quad . \quad (3a)$$

4. Energy reflected (4th) by A_1 and striking dA_2,

$$dq_{r4} = \frac{\sigma e_1}{\pi} \frac{(1 - e_2)^2}{\pi^2} \frac{(1 - e_1)^2}{\pi^2} \int_{A_1} \left\{\frac{\cos \phi_1 \cos \phi_2}{r^2}\right\} \left\{\int_{A_2} \left[\frac{\cos \phi_1 \cos \phi_2}{r^2}\right] \left[\begin{matrix}\text{Second bracketed} \\ \text{term from } 3a\end{matrix}\right] dA_2\right\} dA_1 \cdot dA_2 \quad . \quad . \quad (4a)$$

Then the total energy leaving A_1 and striking A_2 is,

$$q = \frac{\sigma e_1}{\pi} \int_{A_2} (\text{Eq. } 1a + \text{Eq. } 3a + \text{Eq. } 4a +)dA_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5a)$$

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$$= \frac{\sigma e_1}{\pi}\int_{A_2}(a + kb + k^2c + k^3d \ldots .)dA_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6a)$$

$$\text{where, } k = \frac{(1-e_1)(1-e_2)}{\pi^2} \text{ and } a = \int_{A_1}\frac{T_1{}^4 \cos\phi_1 \cos\phi_2 dA_1}{r^2} \quad . \quad . \quad . \quad . \quad . \quad (7a\text{–}8a)$$

$$b = \int_{A_1}\left[\frac{\cos\phi_1\cos\phi_2}{r^2}\right]\left[\int_{A_2}\left(\frac{\cos\phi_1\cos\phi_2}{r^2}\right)\left(\text{Term } a\right)dA_2\right]dA_1 \quad . \quad . \quad . \quad . \quad (9a)$$

B) Energy leaving dA_1 and striking A_2 (same conditions as part A above).

1. Energy emitted dA_1 and striking dA_2,

$$dq = \frac{\sigma e_1 T_1{}^4}{\pi}\frac{\cos\phi_1\cos\phi_2}{r^2}\,dA_1dA_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10a)$$

2. Reflected (1st) by A_2 and striking dA_1,

$$dq = \frac{\sigma e_1 T_1{}^4}{\pi}\frac{(1-e_2)}{\pi}\int_{A_2}\left(\frac{\cos\phi_1\cos\phi_2}{r^2}\right)\left(\frac{\cos\phi_1\cos\phi_2}{r^2}\right)dA_2\cdot dA_1\cdot dA_1 \quad . \quad . \quad . \quad (11a)$$

3. Reflected (2nd) by A_1 and striking dA_2,

$$dq = \frac{\sigma e_1 T_1{}^4}{\pi}\frac{(1-e_1)}{\pi}\frac{(1-e_2)}{\pi}\int_{A_1}\left[\frac{\cos\phi_1\cos\phi_2}{r^2}\right]\left[\int_{A_2}\left(\frac{\cos\phi_1\cos\phi_2}{r^2}\right)\left(\frac{\cos\phi_1\cos\phi_2}{r^2}\right)dA_2\right]dA_1\cdot dA_1\cdot dA_2 \quad . \quad . \quad (12a)$$

4. Reflected (4th) by A_1 and striking dA_2,

$$dq = \frac{\sigma e_1 T_1{}^4}{\pi}\left(\frac{1-e_1}{\pi}\right)^2\left(\frac{1-e_2}{\pi}\right)^2\int_{A_1}\left\{\frac{\cos\phi_1\cos\phi_2}{r^2}\right\}\left\{\int_{A_2}\left(\frac{\cos\phi_1\cos\phi_2}{r^2}\right)\left(\begin{matrix}\text{Second bracket}\\ \text{from}\\ \text{Equation (12}a\text{)}\end{matrix}\right)dA_2\right\}dA_1\cdot dA_1\cdot dA_2 \quad . \quad . \quad (13a)$$

The total energy from dA_1 to A_2 is then,

$$dq = \int_{A_2}(\text{Eq. } 10a + \text{Eq. } 12a + \text{Eq. } 13a + \ldots .) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14a)$$

$$= \frac{\sigma e_1 T_1{}^4}{\pi}\int_{A_2}(a^1 + kb^1 + k^2c^1 + k^3d^1 + \ldots .)dA_2\cdot dA_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15a)$$

$$\text{where } k = \left(\frac{1-e_1}{\pi}\right)\left(\frac{1-e_2}{\pi}\right)$$

$$a' = \frac{\cos\phi_1\cos\phi_2}{r^2}$$

$$b' = \int_{A_1}\left[\frac{\cos\phi_1\cos\phi_2}{r^2}\right]\left[\int_{A_2}\left(\frac{\cos\phi_1\cos\phi_2}{r^2}\right)\left(\text{Term } a'\right)dA_2\right]dA_1$$

DISCUSSION

PRESIDENT FLEISHER: The importance of panel heating may be great or it may not, depending entirely on the point of view. Nevertheless, we can say this for it, that it is receiving more publicity today than practically any other method of heating. Professor Raber stated in his presentation that the idea of panel heating emanated back in the first century and that it is just coming into its own again.

F. C. HOUGHTEN: This is a very important paper on a very timely subject. As

pointed out by President Fleisher, panel heating appears to be about to come into its own. This is a type of paper which frequently comes out of a research laboratory containing a wealth of mathematical data and analysis and which should be considered as a means to an end and not an end in itself.

Certainly if it were necessary to use all the formulae in this paper in order to design panel heating system there would be few panel heating systems installed. A simplification will result from such research and mathematical analysis, and as a final result design data will appear in THE GUIDE for designers and installers of systems throughout the country.

The paper seems to indicate considerable complication in panel heating systems as compared with conventional radiator or air circulating jobs. It appears from work being done in the Research Laboratory in Pittsburgh that the magnitude of these complicating factors is not as great as appears. The factor which makes panel heating seem complicated as compared with the radiator or air circulating system is the effect of radiation from the panel. We must not forget that a radiator job is also a panel heating job to a lesser extent, and even an air circulating job warms up the walls and other surfaces in the room so that even in this case some radiation is obtained.

From work carried out at the Laboratory, it appears that possible variations in the mean radiant temperature found in any practical heating application are not as great as are sometimes supposed.

Two rooms were built recently at the Laboratory, one of which is heated entirely with air circulation and the other by ceiling panels. A year ago one room was heated by radiators. In the room with the air circulating system the mean radiant temperature sometimes goes as low as four or five degrees below the room air temperature, while in a panel heated room the mean radiant temperature may go as much as four or five degrees above the room air temperature in very cold weather. Now this is not a very great spread. Further, if you look at Fig. 1 giving the relationship between mean radiant temperature, air temperature and comfort, you will see that the effect on comfort of the mean radiant temperature is not as great as the effect of the temperature of the air.

The results of the Pittsburgh studies are not included, since they were published only recently. These Pittsburgh results indicate that the curve in Fig. 1 should be even flatter than it is. In other words, the effect of mean radiant temperature on a person's feeling of warmth is not as great as the effect of the air temperature. Stating this in another way, if the mean radiant temperature is raised one degree it will be necessary to depress the dry-bulb temperature of the air considerably less than one degree to give the same feeling of warmth. That is a factor which is going to help in the simplification of the panel heating problem.

Another important factor which will simplify the application of the panel heating system is the fact that the thermostats that control the conditions in the room are not only sensitive to air temperature, but are also sensitive to radiation or the mean radiant temperature within the room. Therefore, even our present thermostatic control systems compensate for the necessary adjustment between mean radiant temperature and air temperature for comfort. If the sensitive element of the thermostat is brought out from the enclosure so as to increase the effect of radiation as compared with convection this compensation effect would be even more pronounced. Undoubtedly, it would be possible to so design the element of a thermostat so that it would integrate this effect fairly well and control the condition in a room so that comfort, with possible variation in mean radiant temperature and air temperature would be obtained.

The study at the Research Laboratory was not designed primarily as a study of panel heating, but essentially as a study to determine the effect of radiation on persons, feeling of warmth as contrasted with the effect of the dry-bulb temperature, the moisture content and the movement of the air. In this study a direct comparison is made of the feeling of warmth of persons in the convection heated room with

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very little radiation, and persons in the panel heated room where there is a considerable elevation in the mean radiant temperature. In a panel heated room the mean radiant temperature was at times as much as 4 deg higher, and in the air circulating room as much as 5 deg lower than the air temperature.

Statistical analysis of a great mass of data shows that a person is warm in the panel heated room at from 2 to 3 deg ET lower air temperature than he is in the convection heated room. A difference of 2 or 3 deg ET is of some importance but not greatly above the threshold of observation because it takes about 3 deg ET to change a person's feeling of warmth from *ideal comfort* to *comfortably warm*. So it is seen that these unusual effects which are important in panel heating are not as great as we frequently imagine. It is, therefore, indicated that the problem of design and installation of panel heating system jobs will not be as great or complicated as sometimes thought. At the present time data are needed for design purposes in THE GUIDE which will give the heat emissions from panels located in the ceiling, side wall or floor for various panel surface temperatures.

S. R. LEWIS: My approach to this question of panel heating leads me to raise some other questions relating to the character of the building construction which Dr. Houghten did not mention. We designed several houses that have been successful in operation for three or four years in which we embedded the hot water coils in the plaster of the ceiling. We used low temperature hot water, never more than 130 deg when 20 deg below zero outside. We were afraid to reduce the amount of the surface and increase the temperature for fear we would get cracks in the plaster. We are now installing three or four jobs and doing a little private research, in which we are using much higher temperatures, but with apprehension as to what may happen to the plaster, whether it will split and crack. We are going to have one or two jobs in which we will use a steel ceiling and will approach steam temperatures in order to determine whether we may reduce the surface of the panels and increase the temperature of the heating medium in the panels. I think the character of the building construction for panel heating should have considerable study to determine how much surface and at how high a temperature that surface may go.

The popularity of panel heating is increasing. For instance you come to a large residence and it is always a difficult task to get the owner, especially the owner's wife, to agree as to where you can put a radiator, because the walls are all windows. If we can put the heat transmitter in the ceiling and do not have any radiators or registers to make it necessary to cut up the carpets, that would be wonderful.

I suppose we have installed and used through this past winter a half-dozen houses in which the bedrooms and corridors are heated in the conventional way, with radiators, but the living room and dining room have panel heating. I have been afraid not to use a separate pump and lower temperature water for the panels than for the radiators but we do not hear any complaints from them and the people like them.

People can live in comfort in houses in which some of the rooms have panel heating and some do not.

F. E. GIESECKE: If a section of the ceiling of this room were equipped with heating coils, it would become the heating panel of the room and would heat the walls and floor so that they, in turn, would also become heating panels for the room.

This room is filled with light radiation, emitted by the electric lamps and reflected and re-reflected by the walls, floor, and ceiling, so that all parts of the room are lighted, although to varying intensities. In exactly the same manner, this entire room is filled with heat radiation, because all portions of the walls, floor, and ceiling are continuously emitting, absorbing, and reflecting heat radiation.

The two important differences between light radiation and heat radiation are that light radiation has a shorter wave length than heat radiation, and that light

radiation enables the human eye to see, whereas heat radiation cannot be sensed by the human eye and is transformed into heat in all bodies by which it is intercepted.

In studying panel warming and cooling, I think frequently of the difficulties which architects encountered in their efforts to design large rooms with good acoustic properties until Professor Sabine, Harvard University, found that the sound waves which were reflected and re-reflected by walls, floor, and ceiling, and which thereby produced unsatisfactory acoustic conditions, could be absorbed by placing proper absorbing material in any convenient part of the room under consideration.

In a similar manner, heat radiation can be produced in a room by placing a heating panel at any convenient point in the room; the heat radiation emitted by such a panel will be reflected and re-reflected by walls, floor, and ceiling, so that the entire room will be filled with heat-radiation, exactly as it is filled with light radiation when light radiation is emitted by one or more lighting units in the room, and as it is filled with sound waves when sound is produced in any part of the room. Of course, in all three cases, the intensity of sound, the intensity of light radiation, and the intensity of heat radiation vary from point to point in the room.

I believe the principal problem in designing heating and cooling panels is not the determination of heat exchange between the human body and a panel placed at some point in the room, but rather the exchange of heat between the human body and all surrounding bodies, whether walls, floor, ceiling, furniture, or other human bodies.

I believe the problem of designing heating and cooling panels is a very broad one and that there is no fundamental difference between the method of heating or cooling buildings by means of panels and that of heating or cooling by means of exposed or concealed radiators, or by means of currents of warm or cool air.

The principal difference between the systems is that with a panel warming system a larger portion of the heat is transmitted by radiation and a smaller portion by convection than is the case when other systems are used.

L. M. K. BOELTER: Mr. Fleisher mentioned the fact that panel heating was developed during the first century. It occurred to me that he might have mentioned the dark ages of panel heating. Any high-ceiling room behaves in a manner almost identical with a panel heated room. The air in the room, however it may be heated, will rise to the ceiling; the ceiling is 15 or 20 deg warmer than the floor, and a large fraction of the interchange between the system and the surroundings will be by radiation. Thus the proponents of residential panel heating are merly adapting a heating system which operated quite satisfactorily in high-ceiling rooms to the present vogue of the low-ceiling room.

I suspect that the larger areas in the comfort curves which were shown by Professor Raber can be accounted for partially by the fact that there is occurring in the transfer mechanism from human bodies another form of radiation which is usually not alluded to in the literature. I refer to gaseous radiation. The carbon dioxide and the water vapor content of the air, the former being negligible, will alter the radiant energy distribution within an enclosed space and roughly stating, about one-fourth of the radiant energy transfer between the body and the surrounding walls will be accounted for by gaseous radiation and the other three-fourths by radiant transfer which obeys the fourth power law. The exponents for gaseous radiation vary anywhere from about the first power up to the third power, depending on the gas concentration, and other variables.

As the comfort curves are analyzed more closely and redeterminations are made greater agreement between the experimenters probably will be found, if the effective gaseous radiation is taken into account. For example, a person perspiring will be surrounded by an interface of air saturated with water vapor. This water vapor will immediately absorb a relatively large fraction of the radiant energy leaving the body and thus will change the distribution of energy between the surrounding air and skin. These comments are tentative, and must be substantiated by experiment.

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PRESIDENT FLEISHER: If you remember the constant energy paper,[3] that was presented at the Kansas City meeting in January, you will have noticed that the lines of equal energy, as represented by constant volume lines, deviated from the effective temperature lines either by being higher or lower than these lines, as indicated by the dry-bulb component at any point on these lines. We contend that this occurred because the investigators who produced the effective temperature lines, particularly in laboratory experiments, had not taken into account the radiant effect of the surrounding walls. You will find that where the Fleishers' lines were in a higher plane than the effective temperature lines, the temperature of the surrounding walls and fixtures was very much lower than the radiant temperature of the body; and that where the Fleishers' lines were in a lower plane than the effective temperature lines, the dry-bulb temperatures or radiant temperatures of the walls and windows were very much higher than the body or surface temperatures. If a calculation of the radiant effect of the walls for various conditions were included in the development of the energy equations, it would probably be found that the coincidence of the energy lines and the effective temperature lines would be exact.

The points that Professor Boelter has brought out bring up a subject of investigation which I have been urging on the Society for years. The work that the Fleishers did in developing the constant energy equation had to do almost entirely with ambient conditions; but as this energy had to be balanced by energy given off by the human body, it is a function of the Society's Research to develop a method of measuring this particular energy given off by the body. As I have stated, we have neglected this side of the question almost entirely. As we are particularly interested in human beings and in their reactions to the ambient conditions and the conditions of the surrounding walls, it is, in my opinion, more important to measure what this body energy factor is.

The whole subject that Professor Boelter has brought up appears to me to be new, certainly as far as the Society is concerned, and is a subject that definitely should be carried further in our Research.

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[3] Comfort and Health and Temperature a Mathematical Solution, by W. L. Fleisher and W. L. Fleisher, Jr. (See p. 175.)

No. 1183

CONCERNING CONSERVATION OF UNDERGROUND WATER WITH SUGGESTIONS FOR CONTROL

By Noel E. Porter,* Sacramento, Calif.

Introduction

THE rapid increase in comfort air conditioning installations presents definite problems to municipalities concerning water supply and disposal. In many large metropolitan areas legislation has been enacted to limit the use and waste of cooling water.

This paper concerns particularly the upper central and northern California valley area where underground water is essential to farming, industry, and domestic supply. Well water temperatures and climatic conditions are such that the economic balance generally favors the performing of a large percentage of cooling requirements directly with water through extended surface coils. With the growth of air conditioning comes an ever increasing demand for cooling water—and the possibility of drawing unlimited quantities from innumerable wells in a concentrated area could effect considerable hazard. It follows that methods for regulation are necessary to the protection and advantage of all concerned.

Well Water and Disposal Facilities

Average well water temperatures in the central valley area centering around Modesto and Stockton are 66-68 F for wells from 90 to 300 ft deep. In the northern valley from Sacramento, north to Chico, water from the 120-140 ft strata is usually 63 F with extreme cases as low as 59 F. Deeper wells yield warmer water, the rise in temperature being from 1 to 2 F per 100 ft of depth below the 140 ft strata.

Test results of a number of wells in Stockton and Sacramento reveal a yield of 10 gpm, or more, per foot of drawdown for an 8 in. diameter well. Water stands at 20 to 30 ft below ground level. The table is subject to a

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Presented at Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, San Francisco, Calif., June, 1941.

drop in late summer, the amount depending upon seasonal rainfall and snowfall, extent of irrigation from wells, etc.

Disposal facilities in the major localities are listed in Table 1. Sanitary systems require pumping at the sewer farms and are usually limited in capacity for cooling waste. Moreover, clear water produces leaks in the joints of certain types of soil pipe. Storm sewer systems drain by gravity and generally have ample capacity for cooling waste needs. The storm sewer systems are limited in coverage however, and generally serve only the metropolitan districts. In the smaller localities, storm sewers are often routed in alternate streets.

To minimize the cost of pumping, nearly every municipality has placed restrictions on cooling waste to the sanitary system, normally a maximum of 5 gpm for each establishment. Where storm sewer systems are eco-

TABLE 1—DISPOSAL FACILITIES IN MAJOR LOCALITIES

CITY	DISPOSAL FACILITIES
Modesto	Dual System[a]
Stockton	Dual System
Lodi	Dual System[b]
Sacramento	Single System[c]
Marysville	Dual System[d]
Chico	Dual System[e]

[a] Separate sanitary and storm sewer systems.
[b] Storm sewer system somewhat limited in coverage for metropolitan area.
[c] Single system for both sanitary and storm disposal.
[d] Storm sewer mains located in alleys.
[e] Storm sewer mains routed in alternate streets.
In general, it is assumed that storm sewer systems have ample capacity for cooling waste needs.

nomically inaccessible, rates up to 15 gpm are sometimes permitted to be discharged to the gutter. The city of Sacramento, largest in the area, is served by a single sewer system, all of which must be pumped to the river. In December of 1937, Sacramento passed an ordinance limiting waste of cooling water to 1 gpm per foot of frontage for each establishment. This ordinance is effectively a conservation ordinance as it definitely limits the amount of water taken from the ground, and not returned, for each establishment. Any water used over the front foot allowance must be disposed to a return well.

Since inception of the Sacramento ordinance, several return well installations have been made. Definite data have not yet been compiled but the results, so far, have been generally satisfactory. Water is usually supplied from the 120-140 ft strata and returned to the 65-80 ft strata. Some return wells have been drilled to the depth of the supply strata (120-140 ft) and the possibility of mixing the warmer water with the supply may be noticeable in the near future. Return wells have proven unsatisfactory in the lower valley. In the city of Fresno, where many disposal wells have been in operation for

some time, there results a very serious rise in the underground water temperature over the summer period.

It is evident that the over use of wells for cooling purposes will in some manner affect the underground supply. There is the possibility of effecting a rise in the water temperature, creating sufficient suction to admit brackish water resulting with pollution, or lowering the water table. The latter would produce serious economic difficulties and might cause considerable damage to

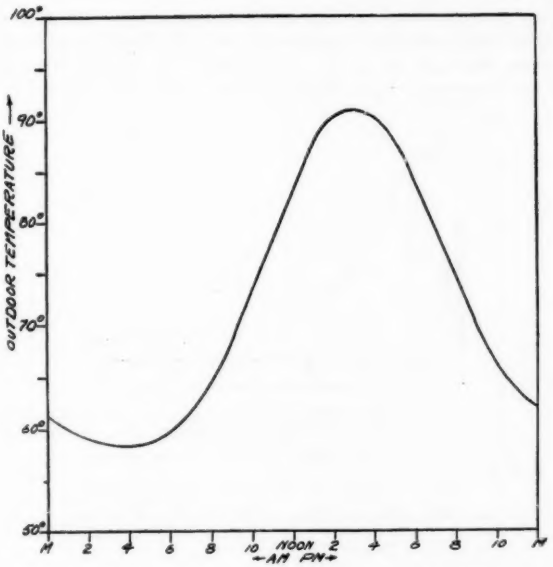

FIG. 1. MEAN TEMPERATURE CURVE FOR THE MONTHS OF JUNE, JULY, AND AUGUST, 1940
(Plotted from data published by U. S. Department of Commerce Weather Bureau, Sacramento, Calif.)

vegetation and trees that are dependent upon the water table for irrigation. Any of these possible results would seriously hazard all users of underground water.

EFFECT OF WELL WATER AND CLIMATE ON DESIGN

Maximum design outdoor temperature for the area is usually taken as 100 deg. The simultaneous maximum design wet-bulb temperature varies gradually from 72 F in the Fresno area down to 67 F in Redding at the northern end of the valley. A typical mean temperature curve for the area is shown in Fig. 1. The curve is plotted using the average of the mean temperatures for the months of June, July, and August, 1940, recorded at

six representative hours of the day and as published by the Sacramento Weather Bureau.

The economically available cool water and prevailing wet-bulb temperatures have developed a trend in commercial air conditioning to systems employing large percentages to 100 per cent ventilation air. Internal latent heat load can often be removed simply by providing sufficient ventilation air. Conditioning apparatus is then required to do sensible cooling only, and water cooling may be used to the utmost economic advantage.

Fresh air of 100 per cent is desirable in hotels and office buildings to minimize the transfer of odors and air-borne bacteria. Often, the additional cost for coils and pumping equipment for *all fresh air* is economically advisable when the cost of an adequate return air system is considered. In cities that are principally agricultural centers and free from industrial soot,

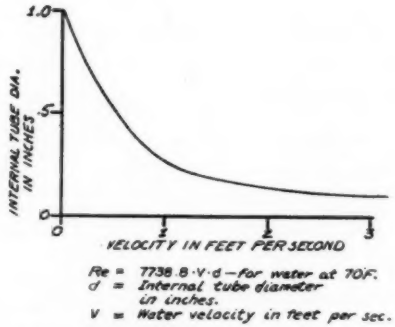

FIG. 2. PLOT OF INTERNAL TUBE DIAMETER AND WATER VELOCITY FOR WHICH REYNOLDS' NUMBER IS 2000

the filter maintenance problem is reduced to a minimum when conditioners are located on the roofs of buildings. The sales advantages of *all fresh air* are evident.

Concordant with the foregoing is the fact that the cooling load for the conditioning apparatus varies in a direct relation with the indoor and outdoor temperature difference, or the outdoor temperature. It follows, that for systems employing a considerable proportion of direct water cooling, the water requirements vary appreciably as the outdoor temperature varies.

Generally, air conditioning systems are operated at full capacity when cooling is required, *i.e.*, as far as direct water cooling is concerned. When refrigeration is employed for further cooling, this part of the system is allowed to cycle thermostatically to provide temperature control, while the water part of the system is operated at maximum flow continuously without regard to cooling requirements. The refrigeration part of the system usually requires a much less quantity of water than the maximum design flow for the system. Inspection of the mean temperature curve indicates that for the greater part of the time that cooling is required, the outdoor temperature is

less than maximum. At such times, a less than maximum flow should suffice.

If attention is given to means for regulating the flow of cooling water in accordance with load, a very substantial saving in seasonal water would result. This is true conservation, as the main concern is the quantity of water taken from the ground over a period of time while the hourly or daily maximum demand is of little importance.

WATER COOLING UNDER VARYING LOAD CONDITIONS

Fig. 2 is a plot of internal tube diameters as a function of water velocity for which Reynolds' Number is 2000; when Reynolds' number is 2000, or

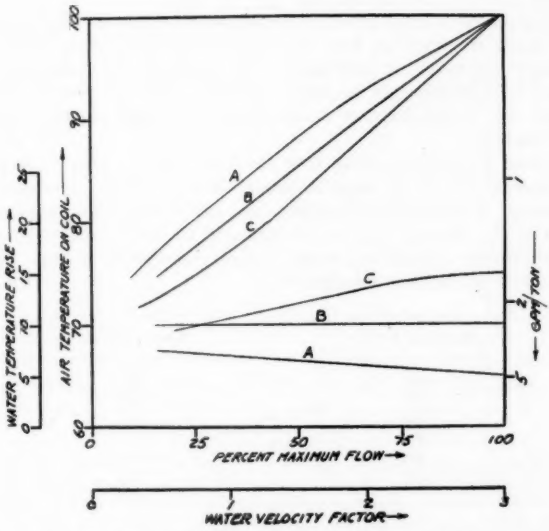

FIG. 3. DRY SURFACE WATER COIL CHARACTERISTICS UNDER VARIOUS OPERATING CONDITIONS

over, flow is turbulent and conventional heat transfer laws apply. It will be noted that for ½ in. to ¾ in. tubes, flow is turbulent for water velocities above ½ fps. Economic coil design for cooling with water at temperatures above the dew-point of the entering air requires a water velocity, at design maximum flow, in the order of 2.5 to 4 fps. It follows, that for normal application, the water velocity, *i.e.* flow, may be safely varied from design maximum down to 20 per cent maximum.

The curves of Fig. 3 show the per cent of design maximum water flow necessary to maintain a fixed delivery temperature at various entering air temperatures. Design conditions are 65 F entering water and a 5 F approach with entering air at 100 F. *Curves A, B,* and *C,* are for a design water tem-

perature rise (hereafter indicated by DWTR) of 5 F, 10 F, and 15 F, respectively. The well-known formula is used:

$$H = U \cdot MTD \cdot A \text{ generally written,}$$
$$H = \text{Area} \times \text{Rows depth} \times HTF \times MTD$$

where HTF [1] (heat transfer factor) is expressed in Btu per square foot per degree MTD per row. The rate of change of HTF with respect to flow is nearly constant over a wide range of face velocities so the curves are typical for a wide choice of physical coil dimensions.

It is interesting to note that *Curve B* (Fig. 3), for a 10 F design water temperature rise, is a straight line over the operating range, *i.e.*, from 100 per cent flow to 20 per cent flow. Over the operating range, the flow drops off faster as the DWTR is reduced, and vice versa.

Curves plotted for a fixed DWTR at different values of design approach are similar in shape to those in Fig. 3. In this case, the flow drops off faster as the design approach is increased. In general, the variation from the straight line relationship will be small so that it may be stated that the water requirements for fixed delivery temperature are approximately proportional to the coil loading.

At this point it is well to note that for most conditioning systems,[2] a fixed delivery temperature will at least maintain desired indoor to outdoor temperature difference. For most applications, and neglecting latent cooling,[3] if only 25 per cent of the internal load is proportional to the indoor and outdoor temperature difference, a fixed apparatus delivery temperature will maintain the following indoor to outdoor relation: [4]

OUTDOOR D. B.	INDOOR D. B.
100	80
95	79
90	78
85	77
80	76
75	75

If more than 25 per cent of the internal load is proportional to indoor and outdoor temperature difference, the delivery temperature can be raised as the entering air temperature drops, and result with a faster falling off of the flow.

For systems using extensive amounts of water cooling, *i.e.*, designed for a close approach, and operating to furnish continuous air delivery with relatively high percentages of ventilation air (75-100 per cent), little cooling effect is gained by maintaining continuous water flow without regard to actual load, or, outdoor temperature. Substantial savings in total water requirements can be expected by modulating the water flow in accordance with load requirements.

The rate at which the coil water flow may be varied with respect to outdoor

[1] HTF values used in calculating are as published by the Trane Co. for type *R* and *S* coils.
[2] Pertinent to this area.
[3] Loc. Cit. Note 2.
[4] HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1941, p. 62.

temperature is largely dependent on the percentage of ventilation air employed, the rate being highest at 100 per cent ventilation air.

Another consideration, which applies particularly to combination water and refrigeration systems, is minimum condenser requirements.[5] This would govern the minimum flow to which the water would normally be reduced, and would depend chiefly upon the percentages of design maximum load attributed

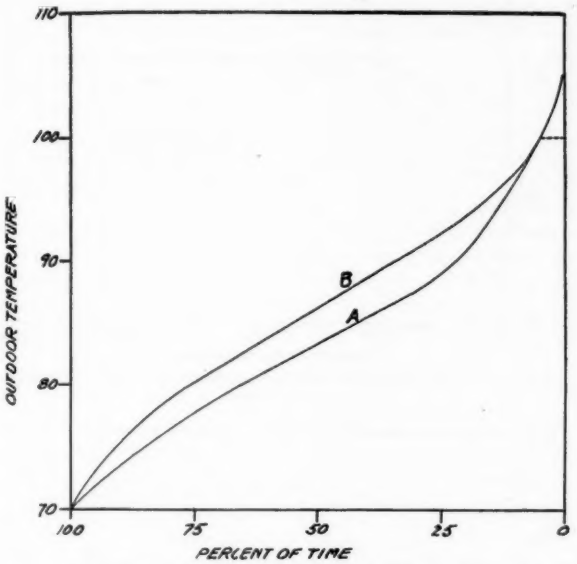

FIG. 4. TOTAL PER CENT OF TIME AT TEMPERATUES ABOVE 70 F FOR MONTHS OF JUNE, JULY, AND AUGUST, 1940, SACRAMENTO, CALIF.

Curve A. Based on average mean daily period from 9:30 a.m. to 9:00 p.m.
Curve B. Based on average mean daily period from 9:30 a.m. to 6:00 p.m.

to direct water cooling and mechanical refrigeration. For example, if the load ratio were 3 to 1 (water to refrigeration) and water cooling was effected at the rate of 2.4 gpm per ton (10 F DWTR), then if the design water temperature is 65 F and minimum condenser requirements 1.2 gpm per ton [6] with 75 F water, minimum percentage of maximum flow to which the water could be reduced would be:

$$\frac{1.2 \times 1}{2.4 \times 3} \cdot 100 = 16.66 \text{ per cent}$$

Fig. 4, *Curve A*, represents the total time for which given temperatures above

[5] It is assumed, in the light of conservation, that the water coil and condenser are series connected.
[6] HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1940, p. 498.

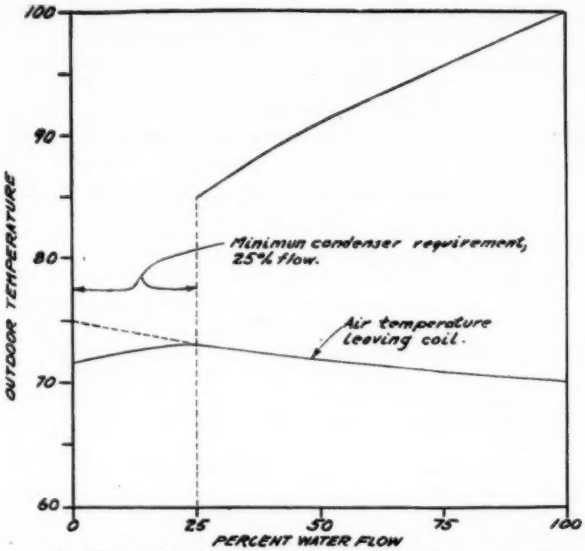

FIG. 5. WATER FLOW, OUTDOOR TEMPERATURE CURVE FOR TYPICAL SYSTEM

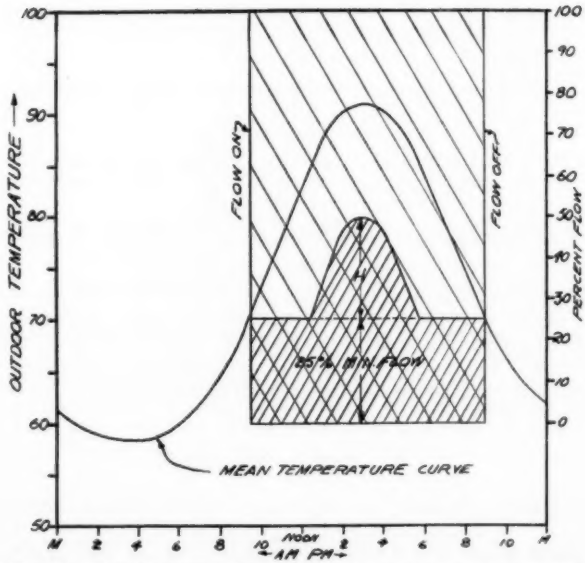

FIG. 6. COMPARATIVE WATER FLOW ANALYSIS FOR SYSTEM OF FIG. 5

70 F are exceeded with respect to the total time above 70 F, expressed as per cent of time above 70 F. The curve is based on the average mean hours of 9:30 a.m. to 9:00 p.m. for the period noted (92 days $\times$ 11.5 hours = 1057 hours above 70 F). The curve is plotted from the mean temperature curve (Fig. 1) and rechecked from actual weather bureau records.[7] The curve is helpful in calculating the total water requirements for the period; 70 F is chosen as the base from the assumption that when the outdoor temperature is below this value ventilation alone will supply cooling or minimum flow only will be required. With this curve and a system water flow characteristic curve (such as Fig. 3) with a vertical line connecting the curve and the horizontal axis at the point of minimum flow, as shown in Fig. 5, the total water requirement for modulated flow may be estimated. For practical purposes, the curves may be divided into increments of Δ per cent flow and Δ per cent time respectively. A summation of the products of the two increments will be the water requirements for the period.[8]

An example is shown in Figs. 5 and 6. Fig. 5 is a per cent water flow curve for a typical system[9] as noted. Fig. 6 is a graphic check on total water requirement by plotting on the mean temperature curve and using an arbitrary per cent water flow scale. For this example, the system operating under modulated water flow requires but 32 per cent of the water required over that if the system is operated at continuous flow whenever the outdoor temperature is above 70 F.

Curve B of Fig. 4 is per cent of time above 70 F for the average mean hours of 9:30 a.m. to 6:00 p.m. and applies to systems that are normally shut down at the close of the business day. The total operating hours for any increment of temperature above 70 F are found by multiplying the corresponding increment of per cent time by the total number of average mean hours for the period.

COMPARSION OF WELL WITH CITY WATER

Pertinent data on city water for the major localities are listed in Table 2. For well systems, the main temperature is substantially constant throughout the year while for surface supply systems, the main temperature is subject to a rise as the summer season progresses.[10] City wells are carefully engineered and distributed so as to draw water over a relatively large area. From the standpoint of conservation, the use of city water is preferred to that from private wells where water rate, temperature, and disposal facilities are reasonably favorable.

The economic balance between well and city water is often in the direction of the latter when such factors as the cost of pumping, cost of well and pump, and proper water control with respect to cooling load, are duly appreciated.

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[7] For the months of June, July, and August, 1940.
[8] Similar to operating cost analysis, HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1941, pp. 471-472.
[9] Very typical in this area.
[10] These points are substantiated in the Department of Commerce reports *Market Research Series*, Nos. 16, 17 for 1937 and 1938. Sacramento water main temperature does not agree with that published in the HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1941, Chapter 26.

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TABLE 2—SOURCE OF CITY WATER SUPPLY

(Temperature at Supply—Average Maximum Water Temperature in Mains and Approximate Commercial Rate for Principal Municipalities in the Area)

CITY	SOURCE	TEMPERATURE	MAIN TEMPERATURE	RATE PER 100[a] CU FT
Modesto	Wells	68	69	$0.076
Stockton	Wells	69	70	$0.075
Lodi	Wells	66	67	$0.050
Sacramento	Surface[b]	—	72–77	$0.080
Marysville	Wells	66.5	67[c]	$0.075
Gridley	Wells	66.5	67	$0.075
Chico	Wells	64	64	$0.071

[a] Rate is expressed in cost per 100 cu ft and the average for a monthly consumption of 180,000 gal. The cost of the first 2000 cu ft is not included and considered normal domestic consumption.
[b] Sacramento is supplied by water from the Sacramento River which is subject to a wide variation in temperature throughout the year.
[c] Applies also to Yuba City.

The effective water rate, ER, corrected for cost of pumping may be written, where AR is the actual rate and rates are expressed in cost per 1000 gal [11]:

$$ER = AR - \frac{\text{head in feet} \times 0.00315 \times C}{\text{pump eff.} \times \text{motor eff.}}$$

where C is cost of power per KWH.

If modulated water flow with respect to load conditions is accomplished (as in examples of Figs. 5 and 6) the effective rate can be further reduced to values as low as 25 per cent of the actual rate, when compared to continuous maximum flow as would normally be the case for a well and pump installation.

Example: Fig. 5 is a water flow-outdoor temperature curve for a typical system with the following conditions: 100 per cent fresh air to be cooled at maximum design outdoor temperature of 100 F air to 60 F and to maintain a corresponding indoor temperature of 80 F. Water cooling effects 75 per cent of maximum load, i.e., from 100 F to 70 F using 2.4 gpm per ton. Air is then cooled at maximum load from 70 F to 60 F by refrigeration requiring 1.8 gpm per ton, or 25 per cent of maximum flow. Fifty per cent of the internal load is proportional to indoor-outdoor temperature difference and an assumption is made that the refrigeration part of the system will effect 10 F cooling at all operating conditions, the resultant air schedule:

OUTDOOR D. B.	INDOOR D. B.	OFF WATER COILS	ENTERING AIR	PER CENT FLOW
100	80	70	60	100
95	79	71	61	73
90	78	72	62	45
85	77	73	63	25
80	76	74	64	25
75	75	75	65	25

Latent cooling load is neglected, or assumed to be absorbed by the refrigeration part of the system.

[11] Cameron Hydraulic Data, 9th Edition.

It can be seen that for temperatures below 85 F, 25 per cent flow is sufficient water. From the per cent of time curve (Fig. 4, *Curve* A) the outdoor temperature is at or below 85 F for 57.5 per cent of the total time above 70 F. Total time flow for this period is then 25 per cent flow × 57.5 per cent time, or a dimensionless value of 1437. (Minimum flow at times when the outdoor temperature is below 70 F is neglected.) The other 42.5 per cent of the total time, the outdoor temperature is between 85 F and 100 F. For approximation, both curves are assumed to be straight lines and the average flow for this range is 57.5 per cent, and total time-flow 57.5 × 42.5 = 2444. Total time flow for period is then 2444 plus 1437 or 3881. This figure, compared to total time flow if the water were operated at maximum flow at total time above 70 F (100 per cent flow × 100 per cent time = 10,000) shows that the system requires but 38.8 per cent of the total water. Using smaller increments in temperature above 85 F would show a greater saving due to the abrupt rise in the per cent of time curve at higher outdoor temperatures.

Example: Fig. 6 is a graphic check on the described system; by plotting values of per cent flow at given outdoor temperatures on the mean temperature curve (Fig. 1), an arbitrary per cent flow scale is set up on the vertical axis. The large lined area (lined 30 F to left) represents the total time-flow if maximum flow were continuous at all time above 70 F as previously described; 100 per cent flow times average mean time of 11.5 hours (9:30 a.m. to 9:00 p.m.) or a dimensionless quantity of 1150. The smaller lined area (lined 30 F to right) is the time-flow for the system when flow is modulated. If the small hump above the 25 per cent flow line is assumed to be a sine curve, the area is easily integrated and becomes (2/pi) times the maximum height (25 per cent flow) times the base (5.5 mean hours) or 87. This added to 25 per cent flow for 11.5 hours gives a total time flow of 375. Compared to 1150, the total water requirement when the system is modulated is 33 per cent as compared to 38.8 per cent indicated in *Example* for Fig. 5.

MECHANICAL METHODS FOR CONTROL

The mechanical problem involved for modulating water flow in accordance with load requirements is one of providing a suitable motor operated water valve having a *modulating* or proportioning type of motor operator interconnected to an expedient thermostatic controller. The controller element can be located in the air stream leaving the coil and the circuit arranged so that the valve will throttle to positions that will admit just sufficient water necessary to maintain the desired leaving air temperature, or the controller element can be of the differential type and located in the air stream entering coil with the circuit arranged so that the valve will operate from the minimum open to fully open position as the entering air temperature varies from design minimum to maximum, and vice versa. Compensated control can be included to provide nearly any desired leaving-entering air temperature schedule.

Attention must be given to the valve port design, pressure drop at various positions, etc., so that the characteristics are suitable to the pressure variations in the water supply and cooling apparatus. If a pump is used for the source of supply, it must have characteristics suitable for operation over the required variations in discharge pressure, non-overloading. The problems are many and not discussed here; however, various types and makes of control equipment that will perform the required functions are commercially available.

LEGISLATIVE METHODS FOR CONTROL

Limiting waste of cooling water to sewers is seemingly an effective method for controlling the underground supply within city limits, *i.e.* for an area supplied with a municipal sewer system. Like the Sacramento ordinance, division

of limitation on a front of footage basis is deemed fair to all as to a certain extent average building size, relative taxation returns, etc., are taken into account.

There are, of course, many buildings with a small front footage to volume or cooling load ratio. In such cases, where the allowable disposal rate is insufficient for cooling needs, other heat removal devices such as return wells, cooling towers, or evaporative condensers are required. It is suggested that disposal might be limited on a square foot of property basis or developed property taxation basis; these bases would tend to effect a better balance and allow greater disposal for the larger buildings, not so dependent on a single factor such as front footage.

Another suggestion would be to limit disposal on a basis using either footage or taxation, with consideration for rate over a period of time, that is, limitation on an average hourly basis so that, with proper water control, greater quantities could be used at the hottest hours while the average over a period of time would be within reasonable limits. This would be true conservation from a supply water standard [12] and minimize the necessity for return wells. However, sewer capacity would become a deciding factor as all maximum flows from cooling systems would occur at the same time.

In some localities, legislation governing the installation of return wells may be required to prevent mixing between the supply and return strata, rise in well water temperature. Specifications as to depth, casing material, etc., can be included in ordinances granting permission to dig wells under city sidewalks.

The possible increase in first cost, and/or operating cost that is effected for systems where restricting ordinances are enforced is in the true sense a small item compared to the probable results of allowing every commercial establishment to draw unlimited quantities of well water continuously.

CONCLUSIONS

The growth of air conditioning in the upper central and northern part of California presents problems to municipalities concerning the use and waste of cooling water. Means for regulation are desired for the protection of the underground supply, particularly in commercial areas.

Climatic conditions are such that a large percentage of the heat to be removed from buildings is in the form of sensible heat. The available cool water is used directly for cooling to a decided economic advantage and makes possible the use of high percentages of ventilation air.

The water rate for the cooling apparatus can be regulated in proportion to cooling load with the result of a substantial saving in total water required for the summer season. True conservation is then accomplished and in many cases the use of city water is warranted.

An effective method for controlling the underground supply in city areas is by providing ordinances limiting the rate of disposal to sewers. Limitation on a foot of frontage basis is used but other methods are suggested and require further study to provide ordinances which more equally distribute the use of water in accordance with cooling needs. Ordinances governing the use of disposal wells may be required in the near future.

[12] See page 313.

The problems are different for each community and necessitate a comprehensive survey of the underground supply, city water facilities, disposal facilities, etc., to secure an ordinance which is best suited and fair to all.

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DISCUSSION

PRESIDENT FLEISHER: We have all had experience with well water troubles. In the city of Brooklyn, N. Y., when I first started to use well water back in the early days, the temperature ran about 52 F and continued at this temperature for many years. Finally when air conditioning came into vogue and it was known there was a gravel strata running through all of Brooklyn and Long Island, everyone attempted to use that water and the city became frightened and insisted upon the water being returned through a return well of approximately the same depth as the supply well. In the course of five years the well water temperature, in the main section of the downtown district, rose from 54 deg to 68 deg and consequently the well water became of no significance at all in the air conditioning. There were several jobs I know of where they had to supply refrigeration where well water had originally been satisfactory.

A. C. BUENSOD: Referring to President Fleisher's incident in Brooklyn, we had a similar experience in Manhattan. We put a little job in the first thing we knew we had 68 deg water. That was food for the refrigeration business. What I am wondering is, in this good state of California and the West Coast, where do you get all the water from? In New York we have conservation and the conservation means that we cannot use the water, we have to go too far away to get it, so I would like to know how much water you are going to have here and is it going to keep up?

JOHN KELLY: In regard to Mr. Buensod's question of where we are going to get the water, 20 or 30 years ago we used to be able to pump the water with a centrifugal pump; now we are lifting it with deep-well turbines set from 100 to 600 ft. When they go down much farther these dams will not do us much good and we will have to give the country back to the Indians.

B. M. WOODS: The problem of water conservation has to be approached in several ways. As the author was careful to say when he opened the discussion, his proposals apply to the part of California in which the well water temperature, which is approximately the mean annual temperature, is low enough to make well water effective. Further, the paper applies only to those regions in which there is ample water.

Throughout the west in general, water supply is a serious matter. There is usually a deficiency. Also, water here as everywhere, should be thought of as a chemical solution, usually a weak solution. In some cases heavy salinity is so high that the treatment of the water leads to serious problems of disposal of the products. In one case deposit of scale in condensers is serious. To keep condensers clean, with water that has fairly high salt content, is a problem of first importance in the more arid portions of the west.

When the water supply of a region is low, the cooling tower becomes the

obvious solution as a means of conserving the supply. Cooling towers are a matter of interest to a number of you on this account. Recognition of the problem by the Society has lead to cooperative research in cooling towers at the University of California. When water becomes expensive, the cooling tower method is clearly economical.

There are so many questions involved in determining when the cooling tower is the correct solution and when use of well water may be preferred that I must not attempt further discussion of them now.

There is one general question I should like to comment on. Air conditioning systems are only satisfactory when their control is satisfactory. Fortunately, conservation of water and satisfactory operation of a system work together when automatic control are developed. Automatic control is rapidly taking the place it must have ultimately. The most important characteristic of our machine age is the fact that we try to relieve the operator of as many problems as possible by incorporating the skill into the machinery of a system. This means that the operator of an air conditioning system should be given as little as possible to do in running the system. I am very much in sympathy with the use of all electrical devices at hand to supplement the mechanical devices and to cause operation of air conditioning systems with the minimum of attention.

The small air conditioning plant will not be widely applied if it costs too much to operate. This means that automatic control must replace hand attention. In small industrial installations it means the substitution of energy at a cost of 25 to 35 cents per horsepower day for man energy at a cost of $150.

AUTHOR'S CLOSURE: I would like to cite another problem of control in Stockton, Calif., where a modulating valve was placed on the outlet of a standard turbine constant speed pump. As the outside temperature varied from a minimum up to 100 F or the maximum indoor temperature, this valve would move from a wide open to a closed position, by-passing water back into the well. With this arrangement, a small instead of a full sized line valve was used and it seemed to work out satisfactorily. Of course, the usual method would be to use a throttling valve and a thermostat located in the air intake or discharge, operating the valve in accordance with the load requirements. Many methods of control could be used, or combinations employed and schedules could be arranged, but this particular method operated with complete satisfaction.

In regard to Mr. Buensod's question, the water in this territory comes from the high Sierras, which feed the entire valley, and these mountains are snow-capped throughout the year. Not only that, but Dr. Woods tells us about the matter of water conservation in the light of all the dams that are being built, so I believe, we will have plenty of water in this state for many years to come.

Our biggest problem is the matter of dropping the water table and effecting the temperature rise in the concentrated areas, and unless we give some thought to the control problem the same thing will happen in this territory as Mr. Fleisher referred to in New York.

No. 1184

THERMAL CONDUCTIVITY OF WOOD

By J. D. MacLean,* Madison, Wis.

INFORMATION on the thermal conductivity of wood and the influence of the more important variables affecting this property is of special interest from the standpoint of building insulation, the use of wood in connection with refrigeration, and in many other fields where the resistance of wood to heat transfer is a major consideration. This information is also of interest for comparing the thermal insulating properties of the different species of woods and for comparing wood with other insulating materials.

Previous studies on the heat conductivity of wood have been comparatively limited in extent from the standpoint of the number of species tested, variations in density within the same species, and the number of tests made on any given wood. Furthermore, the earlier experiments have been chiefly confined to wood having a relatively small moisture range. Often such investigations have been incidental to experiments on insulating products and fabricated structures such as wall sections.

The heat conductivity of wood is dependent on a number of factors of varying degrees of importance. Some of the more significant variables affecting the rate of heat flow in wood are the following: (1) density of the wood; (2) moisture content of the wood; (3) direction of heat flow with respect to the grain; (4) kind, quantity, and distribution of extractives or chemical substances in the wood, such as gums, tannins, or oils; (5) relative density of springwood and summerwood; (6) proportion of springwood and summerwood in the timber; (7) defects, like checks, knots, and cross grain structure.

The purpose of this paper is to report and discuss the results of a large number of heat conductivity experiments that have been made during the last 5 years at the Forest Products Laboratory[1] to determine the influence of some of the more important variables on thermal conductivity. A number of references to various publications dealing with the subject of heat conductivity through wood and other insulating and building materials are included at the end of this paper. References are also given to publications discussing such insulating problems as the condensation of moisture in the walls of buildings.

Preparation and Selection of Specimens: Specimens used in these experiments were selected from 32 species of wood, comprising both softwoods and hardwoods. A special effort was made to select specimens having different densities and different amounts of moisture within each species. Experiments

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[1] Maintained at Madison, Wis., in cooperation with the University of Wisconsin.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, San Francisco, Calif., June, 1941.

TABLE 1—SUMMARY OF DATA FROM CONDUCTIVITY TESTS ON OVEN-DRY WOOD

SPECIES	NO. OF RUNS IN AVERAGE	AVERAGE SPECIFIC GRAVITY	RANGE OF SPECIFIC GRAVITY	AVERAGE CONDUCTIVITY FROM TEST $= K$	AVERAGE COMPUTED CONDUCTIVITY[a] $= K_1$	PER CENT VARIATION $\left[\frac{(K_1 - K)}{K}\right]100$	DIFFERENCE BETWEEN K_1 and K $(K_1 - K)$	REMARKS
Aspen, bigtooth (Populus grandidentata)	5	0.41	0.40 to 0.42	0.71	0.73	+2.8	+0.02	Heartwood & sapwood
Baldcypress (Taxodium distichum)	5	0.39	0.36 to 0.44	0.75	0.71	−5.3	−0.04	Mostly heartwood
Balsa (Ochroma sp.)	4	0.16	0.11 to 0.21	0.41	0.38	−7.3	−0.03	.....
Basswood, American (Tilia glabra)	7	0.38	0.35 to 0.41	0.69	0.70	+1.4	+0.01	Heartwood & sapwood
Douglas-fir (Pseudotsuga taxifolia)	8	0.46	0.37 to 0.49	0.76	0.80	+5.3	+0.04	Heartwood
Elm, rock (Ulmus thomasii)	1	0.76	.....	1.16	1.22	+5.2	+0.06	Heartwood & sapwood
Fir, white (Abies sp.)	2	0.41	0.40 to 0.42	0.71	0.74	+4.2	+0.03	Heartwood
Hemlock, western (Tsuga heterophylla)	2	0.46	0.44 to 0.49	0.79	0.81	+2.5	+0.02	Heartwood
Larch, western (Larix occidentalis)	3	0.57	0.52 to 0.61	0.94	0.95	+1.1	+0.01	Heartwood
Maple, sugar (Acer saccharum)	5	0.68	0.66 to 0.70	1.13	1.11	−1.8	−0.02	Heartwood & sapwood
Oak, red (Quercus sp.)	5	0.67	0.60 to 0.70	1.19	1.09	−8.4	−0.10	Heartwood & sapwood
Pine, southern yellow (Pinus sp.)	7	0.56	0.52 to 0.64	0.94	0.94	0.0	0.00	Heartwood & sapwood
Pine, white (Pinus sp.)	10	0.40	0.34 to 0.45	0.72	0.72	0.0	0.00	Mostly heartwood
Prima vera (Tabebuia bonnellsmithii)	1	0.47	.....	0.82	0.82	0.0	0.00	Heartwood
Redcedar, western (Thuja plicata)	3	0.34	0.34 to 0.35	0.64	0.64	0.0	0.00	Heartwood
Redwood (Sequoia sempervirens)	8	0.40	0.32 to 0.51	0.74	0.72	−2.7	−0.02	Heartwood
Spruce, Engelmann (Picea-engelmannii)	4	0.34	0.31 to 0.40	0.62	0.64	+3.2	+0.02	Mostly heartwood
Tanguile (Shorea polysperma)	4	0.58	0.57 to 0.59	1.00	0.97	−3.0	−0.03	Heartwood

[a] K_1 is conductivity computed from the equation $K_1 = 1.39S + 0.165$.

were made on a number of species with the moisture content ranging from 0 (oven-dry) to that of wood in the green condition.

Two specimens were, of course, used in each run, each pair being matched as closely as possible by cutting them from the same board. The desired initial moisture content was obtained by storing the specimens at suitable constant temperature and humidity until they reached constant weight.

After the specimens reached constant weight in the conditioning room or oven, they were run through a planer and then cut to the required size and placed in test as quickly as possible, in order to avoid any appreciable change in moisture content. Care was taken to select specimens that were practically free from knots, checks, and other defects except in several runs that were made to investigate the influence of such defects.

The specimens were cut to dimensions of approximately $13\frac{1}{2} \times 13\frac{1}{2}$ in. and the thickness generally ranged from about $\frac{1}{2}$ to $\frac{3}{4}$ in. Thickness measurements were made with a micrometer to 0.001 in. Surface measurements (length and breadth) were made to the nearest 1/32 in.

A few specimens from species such as the southern pines, aspen, baldcypress, Engelmann spruce, and some of the oaks and maples, had a considerable amount of both sapwood and heartwood. The specimens of Douglas-fir, yellow birch, western red-cedar, western larch, western hemlock, redwood, tanguile, prime vera, and white pine were all-heartwood and the specimens from the other species were predominantly heartwood.

Many of the Douglas-fir specimens were cut from large diameter logs which made it possible to obtain specimens with small ring curvature. Several runs were therefore made on specimens of this species to compare the conductivity in the radial and tangential directions.

Test specimens of most of the other woods were cut from timbers of moderate diameter so that the direction of heat flow was partly radial and partly tangential. This combination of heat flow in the radial and tangential directions is, of course, more common in structural lumber than heat flow in either the radial or tangential direction alone. A few tests were also made to study the rate of heat flow in the longitudinal direction. These runs were made on Coast Douglas-fir and red-oak.

Conditions of Test: In most of the experiments the specimens were kept under test for about 24 hours to allow the redistribution of moisture to take place and to make sure that constant temperature conditions were established. In a number of runs, the tests were continued for several days to observe the effect on moisture distribution and to find out whether the conductivity was appreciably changed during the longer period of test. In the latter runs the conductivity was determined at different time intervals varying from 24 to 48 hours during the period of test. In general, instrument readings for computation of conductivity were taken at frequent intervals for a period of about 1 hour or more and the results were averaged. No data were taken until the readings showed that constant conditions had been obtained.

Check runs were made at various times by letting the specimens cool to the same temperature on both sides without removing the test panels from the apparatus. The hot plate was then heated again and the conductivity test was repeated. In these check runs the conductivity determined in the two cases was usually within about 1 per cent variation.

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Most of the experiments were made with temperature differences of 40 to 60 F between the hot and cold plate surfaces. In some runs the conductivity was first determined with a temperature difference ranging between 40 and 60 F, and following this the hot plate temperature was raised to give a temperature difference of 80 to 90 F to investigate whether higher temperature

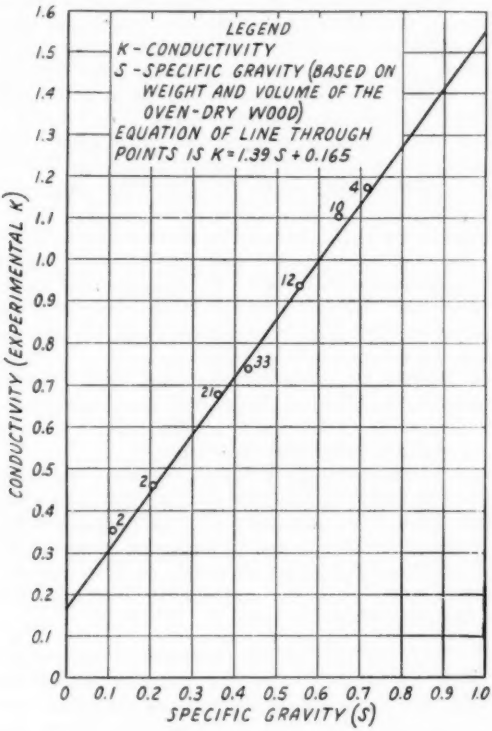

FIG. 1. RELATION BETWEEN CONDUCTIVITY DETERMINED IN EXPERIMENTS ON OVEN-DRY WOOD, AND SPECIFIC GRAVITY OF SPECIMENS TESTED

differences (temperature of hot plate minus temperature of cold plate side) would make any important change in the conductivity. In most of the runs the temperature difference between the two faces of the specimens was about 53 F and the average temperature of the specimens (average temperature of hot plate and cold plate surfaces) was about 85 F.

In this paper the moisture content M is based on oven-dry weight

$$= \left(\frac{W - D}{D}\right) 100,$$

where W is the original weight and D is the weight after oven drying. The specific gravity S was always based on the oven-dry weight and the volume at current moisture content. In this case the oven-dry weight was taken as the original weight of the specimen divided by

$$\left(1 + \frac{M}{100}\right)$$

where M is the percentage of moisture. On this basis the specific gravity

$$S = \left[\frac{W}{\left(1 + \frac{M}{100}\right)}\right] \div (W_w) = \frac{100W}{(100 + M)(W_w)}$$

where W is the original weight of a unit volume of the wood at moisture content M, and W_w is the weight of an equal volume of water. For example, if W is in pounds per cubic foot, W_w is taken as 62.4 lb, the weight of 1 cu ft of water at maximum density.

Studies of the effect on moisture distribution caused by the difference in temperature between the hot and cold plate sides of the specimens were made. These tests comprised about 75 runs of different species having different amounts of water in the wood. The effect on moisture distribution when the wood was kept under test for various periods of time was also studied. In these experiments the following procedure was used in finding the moisture at different distances from the surface of the specimen. Immediately after test, a strip about ½ to ¾ in. wide and 2 or 3 in. long was cut from the central portion of each specimen and thin slices about ⅛ in. or less in thickness were cut from this strip with a mounted slicing knife. The slices were cut parallel with the faces of the test sections. Since the slices were very thin and not smooth enough for accurate measurement of thickness, the average effective thickness was determined from the proportion of the oven-dry weight of the specimen to the total oven-dry weight of all the sliced sections. The average moisture content was determined from moisture sections cut near the pieces which were prepared for slicing.

In engineering practice the conductivity K is commonly taken as the Btu that flow in 1 hour through 1 in. thickness of material 1 sq ft in area, when the temperature difference between the two surfaces is 1 F. This is the value of K shown for these tests and was computed from the formula:

$$K = \frac{Qx}{(T_1 - T_2)A\theta}$$

In this expression Q represents the quantity of heat expressed in Btu.

x = the thickness of the test panels in inches.
T_1 = the temperature on the hot plate side in degrees Fahrenheit.
T_2 = the temperature on the cold plate side in degrees Fahrenheit.
A = the area (expressed in square feet) through which the measured quantity of heat passes.
θ = the time in hours.

Table 1 presents a summary of the data obtained from 84 conductivity tests made on oven-dry specimens of 18 different species. The results of the tests on oven-dry wood showed that the conductivity was, in general, directly pro-

portional to the specific gravity, over the range for the woods tested, as shown in Fig. 1. The plotted data in this curve were averaged for each 0.1 difference in specific gravity and the figures opposite the respective points show the number of runs in each of these group averages. The equation of the line in Fig. 1, showing the relation of conductivity K and specific gravity S for oven-dry wood, is

$$K = 1.39S + 0.165 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

This may also be written, $K = 1.503S + 0.00165P$, where P is the percentage of air space and equals 100 $(1 - 0.685S)$. The constant 0.685 is the specific volume of oven-dry wood substance as determined in helium gas.

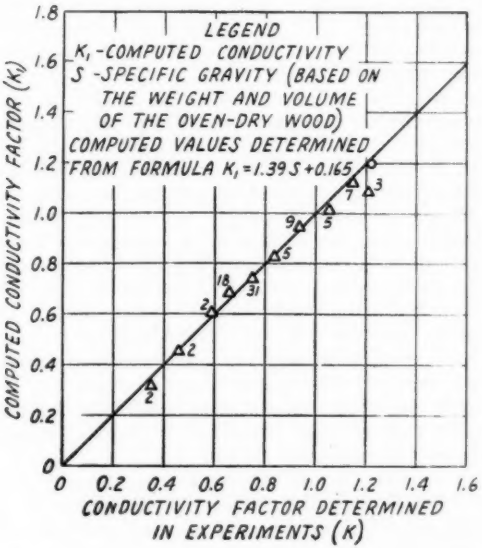

FIG. 2. AVERAGE COMPUTED CONDUCTIVITY PLOTTED AGAINST AVERAGE CONDUCTIVITY DETERMINED IN TESTS ON OVEN-DRY WOOD

(Figures opposite points show number of runs in average. Data are averaged for each 0.10 difference in experimental conductivity factors. Circle shows position of last point when the computed values (K_1) for the three runs on heartwood of red oak had 0.12 added.)

It may be noted that the straight line drawn through the plotted points in Fig. 1 passes through the point where the specific gravity S is zero and the conductivity K is 0.165. This value is the approximate conductivity of air at the average temperature of the test specimens. Fig. 2 shows the average computed conductivity values, using Equation (1), for each 0.1 difference in conductivity, plotted against the corresponding averages of the data deter-

mined in the experiments. The computed values of conductivity are indicated as K_1 in the tables. If the computed and test values were identical, the points would, of course, fall on a 45 deg line passing through 0. The close agreement of the computed and test values is shown by the fact that most of the plotted points fall on or very close to the 45 deg line.

The oak and maple specimens used in these runs contained a considerable proportion of sapwood with the exception of three runs that were made on all heartwood red oak. In these three runs on heartwood the computed conductivity averaged about 0.12 below the test values. This indicates that possibly chemical substances, such as tannin, increased the heartwood conductivity. The addition of 0.12 to the computed value for each of these three runs brings the last point plotted on Fig. 2 practically on the 45 deg line as is shown by the circle.

Species tested in the oven-dry condition had specific gravity values ranging from 0.11 to 0.76. Because of the cellular structure of wood, the relation between specific gravity and conductivity would theoretically be represented by a line gradually curving upward, assuming that the conductivity of wood substance is reasonably constant and the cell structure fairly uniform. The straight line, however, is a very close approximation for the range of specific gravity values between 0 and 0.75. This range of specific gravity, based on oven-dry weight and volume when oven-dry, includes all the native softwoods and practically all the commonly used native hardwoods.

Table 2 shows data obtained in 385 tests on specimens of 32 different woods having varying amounts of water up to the fiber saturation point. In addition, data for 20 runs made on green specimens of 6 different species are shown in Table 3. The formulae for determining the computed conductivities shown in these tables are discussed later.

Changes in Moisture Distribution: Studies of the moisture distribution after test showed that, although the moisture distribution was approximately uniform before test, there were often marked increases in the moisture content of the wood near the cold plate during the test period. These variations in moisture distribution are no doubt due to differences in vapor pressure produced by the differences in temperature. Their range was apparently influenced mainly by the original amount of water in the wood and the magnitude of the differences in temperature between the specimen faces. The rate at which equilibrium was approached was probably affected to a certain extent by the thickness and species.

Table 4 gives data on the moisture content of thin slices of wood about 0.07 to 0.15 in. thick taken from the warm and the cold side, respectively, of a considerable number of test specimens of different woods having various amounts of moisture. These are only part of the total runs made to study moisture distribution. All moisture data are the average values for the two specimens used in each run. Figs. 3 and 4 show representative moisture distribution curves for some of these runs. The data for each curve were obtained by slicing a cross section and determining the moisture content of each slice as previously explained. The results of tests not shown in Table 4 indicate that, when the average moisture content was about 10 per cent or less, the change in moisture distribution was comparatively small.

When the initial moisture content was about that of thoroughly air-seasoned

TABLE 2—SUMMARY OF DATA FROM CONDUCTIVITY TESTS ON SPECIMENS HAVING VARIOUS AMOUNTS OF MOISTURE UP TO FIBER SATURATION POINT

SPECIES	NO. OF RUNS IN AVERAGE	AVERAGE SPECIFIC GRAVITY	RANGE OF SPECIFIC GRAVITY	AVERAGE PER CENT MOISTURE	RANGE OF MOISTURE CONTENT PER CENT	AVERAGE CONDUCTIVITY FROM TEST = K	AVERAGE COMPUTED[a] CONDUCTIVITY = K_1	PER CENT VARIATION $\left[\frac{K_1 - K}{K}\right] 100$	DIFFERENCE BETWEEN K_1 and K $(K_1 - K)$	REMARKS
Ash, white (Fraxinus americana)	7	0.56	0.55–0.57	15.6	12.1–17.2	1.21	1.19	− 1.7	−0.02	Mostly heartwood
Aspen, bigtooth (Populus grandidentata)	10	0.41	0.38–0.47	12.1	9.6–16.4	0.82	0.88	+ 7.3	+0.06	Heartwood & sapwood
Baldcypress (Taxodium distichum)	30	0.38	0.30–0.48	11.7	7.1–15.0	0.86	0.83	− 3.5	−0.03	Mostly heartwood
Balsa (Ochroma sp.)	6	0.17	0.10–0.21	8.0	3.2–10.4	0.47	0.44	− 6.4	−0.03	.....
Basswood, American (Tilia glabra)	20	0.37	0.33–0.41	9.3	3.8–11.5	0.74	0.77	+ 4.1	+0.03	Heartwood & sapwood
Birch, yellow (Betula lutea)	11	0.64	0.62–0.66	10.8	6.8–11.7	1.19	1.24	+ 4.2	+0.05	Heartwood
Douglas-fir (Pseudotsuga taxifolia)	52	0.46	0.41–0.51	18.4	7.2–33.2	0.97	1.03	+ 6.2	+0.06	Heartwood
Elm, rock (Ulmus thomasii)	9	0.65	0.59–0.69	17.9	12.1–31.2	1.34	1.38	+ 3.0	+0.04	Heartwood & sapwood
Greenheart (Ocotea rodioei)	4	0.85	0.84–0.87	18.4	16.5–19.9	1.90	1.79	− 5.8	−0.11	Heartwood
Hemlock, western (Tsuga heterophylla)	5	0.44	0.42–0.46	23.0	13.0–30.0	0.95	1.05	+10.5	+0.10	Heartwood
Larch, western (Larix occidentalis)	13	0.46	0.39–0.64	12.6	11.2–13.4	0.97	0.98	+ 1.0	+0.01	Heartwood
Maple, sugar (Acer saccharum)	27	0.66	0.62–0.72	11.7	5.2–27.8	1.40	1.31	− 6.4	−0.09	Heartwood & sapwood
Oak, red (Quercus sp.)	12	0.62	0.57–0.70	12.4	6.5–19.4	1.35	1.23	− 8.9	−0.12	Heartwood & sapwood
Oak, white (Quercus sp.)	18	0.62	0.55–0.74	11.1	5.4–20.1	1.37	1.23	−10.2	−0.14	Heartwood & sapwood
Pine, southern yellow (Pinus sp.)	59	0.53	0.43–0.71	13.8	5.2–23.8	1.12	1.11	− 0.9	−0.01	Heartwood & sapwood
Pine, white (Pinus sp.)	20	0.36	0.32–0.41	9.8	4.8–17.9	0.77	0.76	− 1.3	−0.01	Mostly heartwood
Prima vera (Tabebuia bonnellsmithii)	9	0.46	0.43–0.48	9.7	1.4–14.7	0.95	0.93	− 2.1	−0.02	Heartwood
Redcedar, western (Thuja plicata)	16	0.32	0.29–0.37	13.3	10.2–20.4	0.69	0.73	+ 5.8	+0.04	Heartwood
Redwood (Sequoia sempervirens)	15	0.39	0.30–0.48	11.7	6.8–12.7	0.82	0.83	+ 1.2	+0.01	Heartwood
Spruce, Engelmann (Picea engelmannii)	12	0.35	0.29–0.39	13.0	12.6–13.2	0.76	0.77	+ 1.3	+0.01	Heartwood & sapwood
Tanguile (Shorea polysperma)	6	0.54	0.53–0.56	10.6	6.0–14.2	1.05	1.08	+ 2.9	+0.03	Heartwood

SPECIES FROM WHICH MATERIAL FOR ONLY ONE OR TWO RUNS WAS AVAILABLE

Beech, American (Fagus grandifolia)	1	0.59	.....	11.1	.....	1.17	1.17	0.0	0.00	Heartwood
Blackgum (Nyssa sylvatica)	1	0.46	.....	10.0	.....	1.00	0.93	− 7.0	−0.07	Heartwood & sapwood
Cherry, black (Prenus serotina)	1	0.68	.....	7.8	.....	1.21	1.26	+ 4.1	+0.05	Heartwood
Chestnut, American (Castanea dentata)	1	0.42	.....	11.0	.....	0.88	0.88	0.0	0.00	Heartwood
Elm, American (Ulmus americana)	1	0.54	.....	9.4	.....	0.98	1.06	+ 8.2	+0.08	Heartwood
Fir, white (Abies sp.)	2	0.38	0.37–0.38	11.7	10.8–12.0	0.78	0.81	+ 3.8	+0.03	Heartwood
Juniper, bigberry (Juniperus utahensis)	1	0.44	.....	16.0	.....	1.13	0.97	−14.2	−0.16	Heartwood & sapwood
Maple, silver (Acer saccharinum)	1	0.47	.....	9.9	.....	1.07	0.95	−11.2	−0.12	Heartwood
Pecan (Hicoria pecan)	1	0.67	.....	10.0	.....	1.38	1.28	− 7.2	−0.10	Heartwood
Sweetgum (Liquidambar styraciflua)	1	0.55	.....	11.0	.....	0.96	1.10	+14.6	+0.14	Heartwood & sapwood
Sycamore, American (Platanus occidentalis)	1	0.52	.....	9.0	.....	1.07	1.02	− 4.7	−0.05	Heartwood

PLYWOOD SPECIMENS

Douglas-fir (Pseudotsuga taxifolia)	12	0.52	0.43–0.56	9.2	4.3–12.0	0.98	1.02	+ 4.1	+0.04	Heartwood

CORRECTION FOR INFILTRATED MATERIALS

When 0.12 is added to compute conductivity K_1 for greenheart, for 14 runs on heartwood specimens of maple, and for all runs on heartwood specimens of red and white oak the following computed values are obtained which differ but little from the conductivity determined in the tests.

Greenheart (Ocotea rodioei)	4	...	.....	...	.....	1.90	1.91	+ 0.5	+0.01	.....
Maple, sugar (Acer saccharium)	27	...	.....	...	.....	1.40	1.37	− 2.1	−0.03	.....
Oak, red (Quercus sp.)	12	...	.....	...	.....	1.35	1.35	0.0	0.0	.....
Oak, white (Quercus sp.)	18	...	.....	...	.....	1.37	1.35	− 1.5	−0.02	.....

[a] K_1 is conductivity computed from the equation $K_1 = S[1.39 + 0.028M] + 0.165$.

TABLE 3—SUMMARY OF DATA OBTAINED IN CON-

SPECIES	NO. OF RUNS IN AVERAGE	AVERAGE SPECIFIC GRAVITY	RANGE OF SPECIFIC GRAVITY
Ash, white (Fraxinus americana)	1	0.48	.....
Fir, white (Abies sp.)	3	0.36	0.33 to 0.38
Maple, sugar (Acer saccharum)	3	0.62	0.61 to 0 63
Oak, red (Quercus sp.)	10	0.56	0.54 to 0.59
Oak, white (Quercus sp.)	1	0.68	.....
Redwood (Sequoia sempervirens)	2	0.36	.....

* K is computed conductivity from equation $K = S\,[1.39 + 0.038\,M] + 0.165$.

wood (12 to 15 per cent) a large part of the redistribution of moisture usually occurred during the first 24 hours of test. This may be noted to be the general case from an examination of the data in Table 4 and is also shown by the moisture distribution curves for different periods of test. A few representative curves are given in Fig. 3. (See runs 90, 93, 99 and 100.) This early redistribution is somewhat obscured for some of the runs listed in Table 4 because the thickness of the slices from near the surfaces was more or less variable. The thinner slices would obviously show a higher moisture content near the cold surface and a lower moisture content near the hot-plate surface because the moisture content given is the average for the slice as taken. In addition, the wood in some specimens was more resistant to moisture movement than it was in others. Since this would affect the rate of moisture movement, the curves of moisture distribution for a given time interval will naturally show a certain amount of variability for different species and also to some extent within the same species.

When the initial moisture content was about 15 per cent or higher, it was noted after test that the wood surface in contact with the cold-plate was often damp or wet whereas that near the hot-plate side was at a moisture content ranging from about 7 to 12 per cent. Under such conditions some moisture naturally adhered to the metal surface of the cold plate. This amount of moisture was probably small but could not be determined.

Only a comparatively small change in moisture distribution occurred in a run on Douglas-fir specimens that had an average moisture content of 16.4 per cent, when the specimens were under test only 5 hours. The curve of moisture distribution for this test is shown for run 92 in Fig. 3. Moisture distribution curves for runs 78 and 157 (Fig. 3) show that, when the initial moisture content was in the range of about 20 to 30 per cent, there was from 20 to 40 per cent difference at the end of the test, between the moisture content of the wood near the cold-plate and that near the hot-plate side. This is also shown for other runs, by data given in Table 4. It should be noted that the moisture content at the surface on the cold-plate side would be somewhat higher than the average shown in Table 4 for the thin section which included that surface.

A large proportion of the runs in these conductivity experiments was made

DUCTIVITY TESTS ON GREEN OR WET SPECIMENS

AVERAGE MOISTURE CONTENT PER CENT	RANGE OF MOISTURE CONTENT PER CENT	AVERAGE CONDUCTIVITY FROM TEST = K	AVERAGE COMPUTED CONDUCTIVITY* THERMAL = K_1	PER CENT VARIATION $\left[\frac{(K_1 - K)}{K} 100\right]$	DIFFERENCE BETWEEN K_1 and K $(K_1 - K)$
91.1	.....	2.60	2.49	−4.2	−0.11
107.7	77.8 to 129.8	2.10	2.13	+1.4	+0.03
50.0	41.0 to 58.6	2.16	2.19	+1.4	+0.03
60.3	36.0 to 82.0	2.23	2.24	+0.9	+0.02
57.6	.....	2.59	2.60	+0.4	+0.01
80.2	71.7 to 88.9	1.89	1.76	−6.9	−0.13

on specimens in which the initial moisture content ranged from a little above zero to about the fiber-saturation point, because data on conductivity are of principal interest in this range. Table 5 shows the number of runs made on each species for moisture content intervals of 5 per cent up to a little above the fiber-saturation point. All runs made on specimens having an average moisture content over 40 per cent are placed in one group and are classed as green.

Effect of Different Periods of Test on Conductivity: Several runs were made to study the effect of different periods of test on the conductivity since it was recognized that the final moisture distribution was dependent to a certain extent on the time the wood was exposed to temperature differences between the two faces. Data on these tests are given in Table 6. The results indicate that when the average moisture content was about that of air-seasoned wood, 15 per cent or less, there was usually no significant change in conductivity after 24 hours in test. In fact, data taken 5 or 6 hours after specimens having this range of moisture content were placed in test, showed there was little change and often no change in conductivity from that found after test periods of 24 hours or more.

When the average moisture content was in the range of 20 to 35 per cent there was some tendency for the conductivity to decrease with increase in time of test. This is shown in data for runs 290, 303 and 306 in Table 6. This change would probably continue until a condition of moisture equilibrium was established for the differences in temperature between the two faces. It is very probable that a considerable part of the redistribution of moisture may have occurred before the first readings of conductivity were taken. For this reason the first determination of conductivity is not necessarily the maximum conductivity.

Green wood having a moisture content of about 60 per cent or more showed a much slower trend toward a lower conductivity than specimens having a lower moisture content close to the fiber-saturation point. This slow change found for the wet wood may be on account of the smaller amount of air space in the wood cells because of the large amount of water present. The moisture distribution curves for runs 254 and 326 (Fig. 4) show that even after 72 to 90 hours in test (3 to 3¾ days) the moisture content of the wood next to the hot plate was not below the fiber-saturation point.

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TABLE 4—COMPARISON OF MOISTURE CONTENT IN WOOD NEAR HOT AND COLD PLATE SURFACES, DETERMINED IN RUNS ON VARIOUS SPECIES

SPECIES	AVERAGE MOISTURE CONTENT OF SPECIMENS	MOISTURE CONTENT IN WOOD NEAR HOT PLATE[a]	MOISTURE CONTENT IN WOOD NEAR COLD PLATE[a]	TIME IN TEST[b]	THICKNESS OF SPECIMENS USED IN TEST	AVERAGE SPECIFIC GRAVITY	TEMPERATURE DIFFERENCE BETWEEN HOT AND COLD PLATE SURFACES F	RUN NUMBER
	PER CENT	PER CENT	PER CENT	DAYS	INCH			
Ash, white (Fraxinus americana)	91.1	53.4	98.3	1	0.820	0.48	52	84
Aspen, bigtooth (Populus grandidentata)	15.0	9.4	24.6	1	0.842	0.40	55	97
Baldcypress (Taxodium distichum)	11.4	8.6	13.8	2	0.615	0.34	52	297
do.	12.2	9.8	16.2	1	0.711	0.39	59	120
do.	13.1	12.7	17.7	1	0.766	0.36	54	122
do.	13.6	9.4	17.6	5	0.760	0.32	53	105
do.	13.6	12.4	16.0	1	0.770	0.31	63	119
do.	14.0	13.4	16.2	1	0.736	0.31	56	111
Balsa (Ochroma sp.)	10.4	4.8	11.6	1	0.777	0.15	64	108
Basswood, American (Tilia glabra)	10.6	8.6	14.1	1	0.638	0.40	53	87
do.	11.5	7.6	11.8	1	0.606	0 39	51	86
Birch, yellow (Betula lutea)	11.2	9.2	11.2	1	0.785	0.63	48	60
do.	11.4	8.8	12.1	1	0.874	0.64	49	59
Douglas-fir (Pseudotsuga taxifolia)	10.2	8.6	10.8	1	0.660	0.47	52	91
do.	12.4	7.2	25.4	6	0.562	0.50	51	288
do.	13.0	9.5	15.2	4	1.495	0.47	53	294
do.	14.0	10.7	17.6	2	0.744	0.51	55	102
do.	14.0	9.3	23.8	5	0.684	0.50	48	101
do.	14.0	9.3	17.4	2	0.631	0.44	49	79
do.	14.0	8.6	15.6	1	0.628	0.45	49	94
do.	14.3	10.8	18.0	1	0.500	0.43	43	90
do.	14.4	8.3	18.1	3	0.608	0.45	50	93
do.	16.4	14.2	16.9	0.208 (5 hours)	0.628	0.46	48	92
do.	20.4	10.8	33.0	1	0.707	0.44	46	71
do.	21.9	12.4	31.8	1	0.709	0.44	46	106
do.	22.5	10.6	48.6	1	0.702	0.44	49	89
do.	27.7	12.6	47.0	1	0.713	0.48	47	77

Douglas-fir (Pseudotsuga taxifolia).	28.8	13.4	48.0	1	0.829	0.46	49	78
do.	34.6	8.0	102.2	3	0.716	0.49	63 and 69	303
do.	34.8	9.4	99.0	3	0.743	0.48	55 and 64	306
Fir, white (Abies sp.)	77.8	37.7	97.4	3	0.800	0.37	77	276
do	115.4	106.8	122.5	3	0.737	0.38	42	258
Maple, sugar (Acer saccharum)	13.0	11.6	13.2	2	0.812	0.68	35	67
do.	13.6	11.4	13.4	1	0.750	0.68	38	69
do.	13.4	11.4	13.4	1	0.775	0.68	40	73
do.	13.7	9.0	19.2	5	0.795	0.69	49	98
do.	13.8	11.0	14.7	1	0.749	0.66	48	99
do.	13.9	12.0	14.0	1	0.810	0.69	40	72
do.	14.1	10.0	19.7	6	0.843	0.66	41	100
Oak, red (Quercus sp.)	58.4	16.6	68.1	4	0.795	0.55	66	285
do.	61.4	34.0	69.8	1	0.727	0.55	45	255
do.	64.2	33.7	82.0	3	0.750	0.56	41	254
do.	82.0	70.4	92.9	3	0.707	0.58	48	326
Pine, southern yellow (Pinus sp.)	11.8	7.6	14.8	1	0.707	0.53	50	152
do.	12.2	10.0	15.1	1	0.629	0.49	38	54
do.	13.1	9.0	15.7	1	0.652	0.49	43	53
do.	14.9	6.6	37.8	6	0.720	0.44	54	161
do.	17.6	11.4	33.0	1	0.808	0.54	47	160
do.	18.0	10.5	21.4	1	0.565	0.45	37	113
do.	18.2	12.6	31.8	1	0.804	0.50	50	157
Prima vera (Tabebuia bonnell-smithii)	7.4	6.7	7.6	1	0.808	0.43	51	104
do.	12.6	9.9	15.8	1	0.828	0.44	52	65
do.	13.3	9.5	13.4	1	0.808	0.46	42	66
do	14.7	8.1	21.8	2	0.822	0.46	63	63
Redcedar, western (Thuja plicata)	12.1	7.6	18.6	1	0.725	0.30	53	57
do.	12.5	7.4	15.1	1	0.794	0.30	47	55
do.	16.4	10.0	27.6	1	0.652	0.31	59	121
do.	19.0	9.1	50.7	1	0.685	0.30	61	118
Spruce, Engelmann (Picea engelmannii)	12.8	6.6	34.2	3	0.692	0.31	66	171

[a] The moisture content values shown for the hot and cold plate sides are the averages found in thin slices varying from about 0.07 to 0.15 inch in thickness. The average moisture content of the specimens was determined from separate moisture sections cut near the sliced specimens.

[b] Each day in test was approximately 24 hours.

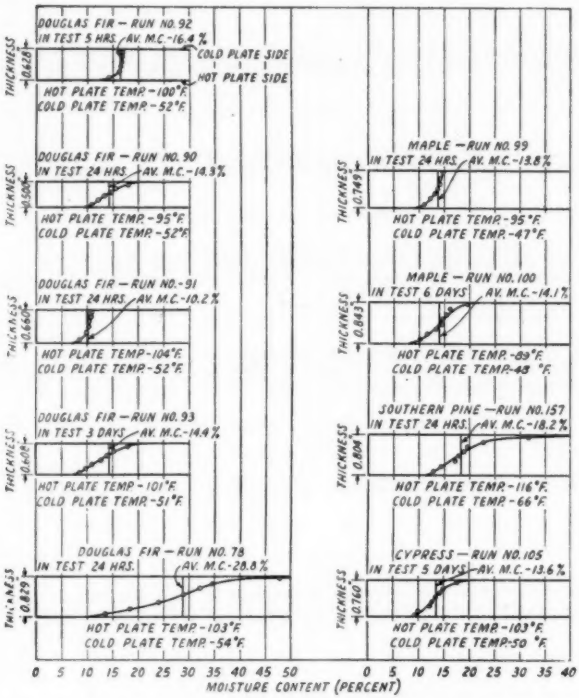
DOUGLAS FIR — RUN NO. 92
IN TEST 5 HRS. AV. M.C.-16.4 %
COLD PLATE SIDE
HOT PLATE SIDE
HOT PLATE TEMP.-100°F.
COLD PLATE TEMP.-52°F.
DOUGLAS FIR — RUN NO. 90
IN TEST 24 HRS. AV. M.C.-14.3%
HOT PLATE TEMP.-95°F.
COLD PLATE TEMP.-52°F.
DOUGLAS FIR — RUN NO.-91
IN TEST 24 HRS. AV. M.C.-10.2 %
HOT PLATE TEMP.-104°F.
COLD PLATE TEMP.-52°F.
DOUGLAS FIR — RUN NO. 93
IN TEST 3 DAYS AV. M.C.-14.4 %
HOT PLATE TEMP.-101°F.
COLD PLATE TEMP.-51°F.
DOUGLAS FIR — RUN NO. 78
IN TEST 24 HRS. AV. M.C.-28.8 %
HOT PLATE TEMP.-103°F.
COLD PLATE TEMP.-54°F.
MAPLE — RUN NO. 99
IN TEST 24 HRS. AV. M.C.-13.8 %
HOT PLATE TEMP.-95°F.
COLD PLATE TEMP.-47°F.
MAPLE — RUN NO. 100
IN TEST 6 DAYS AV. M.C.-14.1 %
HOT PLATE TEMP.-89°F.
COLD PLATE TEMP.-43 °F.
SOUTHERN PINE — RUN NO. 157
IN TEST 24 HRS. AV. M.C.-18.2 %
HOT PLATE TEMP.-116°F.
COLD PLATE TEMP.-66°F.
CYPRESS — RUN NO. 105
IN TEST 5 DAYS AV. M.C.-13.6 %
HOT PLATE TEMP.-103°F.
COLD PLATE TEMP.-50 °F.
THICKNESS
0 5 10 15 20 25 30 35 40 45 50
0 5 10 15 20 25 30 35 40
MOISTURE CONTENT (PERCENT)

FIG. 3. MOISTURE DISTRIBUTION CURVES FOR SEASONED OR PARTIALLY SEASONED WOOD, AFTER DIFFERENT PERIODS IN TEST

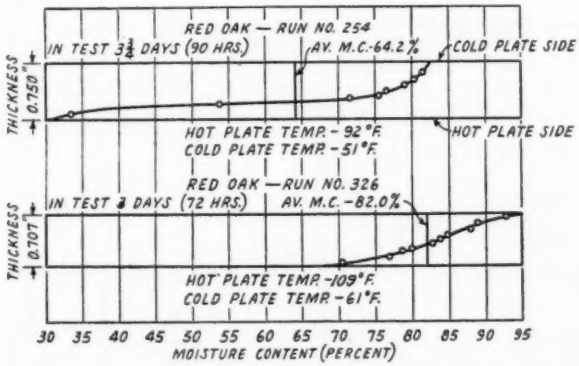
RED OAK — RUN NO. 254
IN TEST 3¾ DAYS (90 HRS.) AV. M.C.-64.2 %
COLD PLATE SIDE
HOT PLATE SIDE
HOT PLATE TEMP.-92°F.
COLD PLATE TEMP.-51°F.
RED OAK — RUN NO. 326
IN TEST 3 DAYS (72 HRS.) AV. M.C.-82.0%
HOT PLATE TEMP.-109°F.
COLD PLATE TEMP.-61°F.
THICKNESS
0.750"
0.707"
30 35 40 45 50 55 60 65 70 75 80 85 90 95
MOISTURE CONTENT (PERCENT)

FIG. 4. MOISTURE DISTRIBUTION CURVES FOR GREEN RED OAK AFTER DIFFERENT PERIODS IN TEST

Results of these tests show that, since the redistribution of moisture has a tendency to reduce rather than increase the conductivity, one can logically assume that the conductivity data will represent maximum values that may be expected for the particular average moisture content, regardless of the time required to establish equilibrium moisture conditions. While it is recognized that the conductivity as determined does not represent the conductivity of the wood under the original conditions of moisture distribution, even if it were possible to devise a method of finding the conductivity without disturbing the original moisture balance, such data would not represent conditions commonly encountered. A moisture gradient necessarily occurs when wood containing moisture is used in any type of structure where a temperature gradient exists between faces. Since specimens tested in the hot-plate apparatus do not evaporate moisture from the cold surface, the average moisture content will not change with change in moisture distribution except for a very slight moisture loss at the edges that does not affect the interior portion. Whether moisture evaporates from the cold side when wood is used commercially depends upon a variety of factors. These are discussed at some length in other publications which deal with condensation problems in buildings.[14, 15, 21, 22, 23]

COMPUTATION OF CONDUCTIVITY

Since both the specific gravity and moisture content can be determined or estimated within reasonable limits for a given species and for particular conditions of service under which the wood is used, a formula for computing the approximate conductivity would be of particular assistance in questions relating to the insulating properties of different woods. An equation showing the relation of conductivity, specific gravity, and moisture content would make it possible to disregard species and to calculate, within reasonable limits, the conductivity of any wood without making conductivity tests.

Published data frequently give the conductivity and weight per cubic foot for a particular species, but no data are given regarding the moisture content at time of test. Since differences in the specific gravity of wood have a very different effect on conductivity from that caused by differences in the amount of water in the wood, it is evident that the weight per unit volume might be the same but the conductivity could vary considerably, depending on the proportional weight of water and wood substance. Although it is not practicable to compute the exact conductivity of wood of given density and moisture content because of the number of variables involved, the results of the experiments discussed in this paper indicate that it is possible to compute the conductivity closely enough for practical purposes.

In deriving an expression for the relation of conductivity, air space, moisture content, and specific gravity, it has been assumed that the conductivity of water-free wood is represented by the formula

$$K = 1.503S + \left(\frac{P}{100}\right)(0.165) = 1.39S + 0.165$$

as shown in Equation (1). In order to find an expression for computing the conductivity of wood containing water it was assumed that another term MSX could be added where MS = the percentage of volume occupied by water

TABLE 5—RANGE OF MOISTURE CONTENT OF TEST SPECIMENS OF DIFFERENT SPECIES

SPECIES	NUMBER OF RUNS MADE AT MOISTURE CONTENT OF								TOTAL NO. OF RUNS ON EACH SPECIES
	Zero or Oven Dry	Between Zero and 5 Per cent, Incl.	Between 5 + and 10 Per cent, Incl.	Between 10 + and 15 Per cent, Incl.	Between 15 + and 20 Per cent, Incl.	Between 20 + and 25 Per cent, Incl.	Between 25 + and 30 Per cent, Incl.	Green or Wet Wood (Over 40 Per cent)	
Ash, white (Fraxinus americana)	...	...	...	1	6	...	...	1	8
Aspen, bigtooth (Populus grandidentata)	5	...	1	8	1	...	...	...	15
Baldcypress (Taxodium distichum)	5	...	6	24	...	...	...	...	35
Balsa (Ochroma sp.)	4	1	4	1	...	...	...	...	10
Basswood, American (Tilia glabra)	7	2	9	9	...	...	...	...	27
Birch, yellow (Betula lutea)	...	...	1	10	...	...	...	...	11
Douglas-fir (Pseudotsuga taxifolia)	8	...	3	27	1	6	15	...	60
Elm, rock (Ulmus thomasii)	1	...	...	5	2	...	2	...	10
Fir, white (Abies sp.)	2	...	...	...	...	...	...	3	5
Greenheart (Ocotea rodioei)	...	...	...	...	4	...	...	...	4
Hemlock, western (Tsuga heterophylla)	2	...	...	1	...	2	2	...	7
Larch, western (Larix occidentalis)	3	...	...	13	...	...	...	...	16
Maple, sugar (Acer saccharum)	5	...	7	19	...	...	1	3	35
Oak, red (Quercus sp.)	5	...	1	10	1	...	...	10	27
Oak, white (Quercus sp.)	...	...	9	5	3	1	...	1	19
Pine, southern yellow (Pinus sp.)	7	...	2	43	7	7	...	...	66
Pine, white (Pinus sp.)	10	2	5	11	2	...	...	...	30
Prima vera (Tabebuia bonnell-smithii)	1	3	1	5	...	...	...	...	10
Redcedar, western (Thuja plicata)	3	...	...	12	3	1	...	...	19
Redwood (Sequoia sempervirens)	8	...	1	14	...	...	...	2	25
Spruce, Englemann (Picea engelmannii)	4	...	...	12	...	...	...	...	16
Tanguile (Shorea polysperma)	4	...	2	4	...	...	...	...	10
Miscellaneous species[a]	...	...	6	5	1	...	...	...	12
PLYWOOD SPECIMENS									
Douglas-fir (Pseudotsuga taxifolia)	...	1	5	6	...	...	...	...	12
Total									489

[a] Species on which only one or two runs were made.

and X = a factor representing the effect of moisture on conductivity. The equation for the conductivity of wood containing water could then be written:

$$K = 1.503S + \frac{P}{100}(0.165) + MSX \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Substituting for $\frac{P}{100}$ its value

$$\left[1 - S\left(0.685 + \frac{M}{100}\right)\right]$$

Equation (2) becomes, $K = S\,[1.39 + M\ (X - 0.00165)] + 0.165$.

If Z represents the term $(X - 0.00165)$

$$K = S[1.39 + MZ] + 0.165. \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

The factor Z will necessarily vary within a certain range, partly because the moisture distribution is not uniform and also because of variability in the wood structure, extractives, and other factors that affect the conductivity of oven-dry wood. For moisture content values below the fiber-saturation point (taken as 30 per cent), the majority of the species tested had a value of Z ranging between 0.018 and 0.039 and the average for the different woods with a moisture content range up to the fiber-saturation point was about 0.028. The average value for Z determined for the tests on specimens having from 50 to over 100 per cent moisture was 0.038.

There is, of course, no strict demarcation for the value of Z at the fiber-saturation point. In general, Z should probably increase somewhat gradually up to a maximum for green material. The equations are necessarily based on average values for both oven-dry wood and for wood containing varying amounts of water. For moisture-content values over 30 but under 40 per cent, it would probably be a little closer to use the factor 0.028 in computing K even though 40 per cent moisture content is about 10 per cent above the fiber-saturation point, taken as 30 per cent.

Substituting 0.028 and 0.038 in Equation (3) for the respective groups:

$$K = S(1.39 + 0.028M) + 0.165 \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

for wood having varying amounts of moisture up to a little over the fiber-saturation point (values under 40 per cent); and

$$K = S(1.39 + 0.038M) + 0.165 \quad \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

for green wood having 40 per cent or more, moisture.

In the foregoing expressions, K = the conductivity, M = the moisture content, and S = the specific gravity as previously defined. It is evident that if the value of $M = 0$ in Equations (4) and (5), both become the same as Equation (1) derived for oven-dry wood.

The difference in the computed and the experimental value of conductivity for partly seasoned wood was highest for heartwood specimens of red and white oak, greenheart, and maple. Three runs made on oven-dry heartwood specimens of red oak and one run on maple specimens, mostly heartwood, also showed somewhat higher conductivities, considering the specific gravity, than the other woods. On the other hand, runs made on heartwood specimens of partly seasoned rock elm, wite ash and yellow birch, which are also relatively heavy woods, did not show the difference in computed and test conductivity values noted for heart material of the other woods named.

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TABLE 6—CONDUCTIVITY OF WOOD AFTER DIFFERENT PERIODS OF HEATING

SPECIES	AVERAGE MOISTURE CONTENT PER CENT	TIME IN TEST HOURS	CONDUCTIVITY K	RUN NO.	REMARKS
Baldcypress (Taxodium distichum)	13.6	24	0.72	105	
		96	0.73		
		120	0.74		
Baldcypress (Taxodium distichum)	11.4	24	0.82	297	
		48	0.82		
Douglas-fir (Pseudotsuga taxifolia)	14.0	24	0.92	101	
		96	0.90		
		120	0.91		
Douglas-fir (Pseudotsuga taxifolia)	14.0	24	0.97	102	
		48	0.95		
Douglas-fir (Pseudotsuga taxifolia)	12.4	2	0.90	288	
		20	0.85		
		44	0.87		
		113	0.85		
		144	0.83		
Douglas-fir (Pseudotsuga taxifolia)	34.6	24	1.11	303	Specimens soaked in water about 6 weeks before test
		72	0.99		
Douglas-fir (Pseudotsuga taxifolia)	34.8	24	1.21	306	
		72	1.02		
Maple, sugar (Acer saccharum)	14.1	24	1.58	100	
		96	1.59		
		120	1.56		
Maple, silver (Acer saccharinum)	9.9	24	1.07	464	
		48	1.06		
Oak, red (Quercus sp.)	64.2	60	2.32	254	
		90	2.25		
Oak, red (Quercus sp.)	66.7	24	2.23	257	
		48	2.24		
Oak, red (Quercus sp.)	58.4	24	2.20	285	
		96	2.10		
Oak, red (Quercus sp.)	82.0	18	2.88	326	Soaked in water about one month before test
		72	2.87		
Pine, southern yellow (Pinus sp.)	14.9	24	1.01	161	
		96	1.01		
		120	1.00		
		144	0.98		
Pine, southern yellow (Pinus sp.)	23.8	24	1.24	290	
		120	1.13		
Redwood (Sequoia sempervirens)	71.7	24	1.66	314	Soaked in water about 6 weeks before test
		48	1.66		

Some of the maple specimens had a considerable amount of sapwood and the values of the computed and experimental conductivity factors for these specimens were fairly close. It appears possible that extractives, such as

tannin in the oaks, and various substances found in the heartwood of other species may be largely responsible for the higher conductivity of the woods in question. The difference between the computed value of K and that found from the tests on the heartwood specimens of oak, greenheart, and maple was fairly constant and was in the neighborhood of 0.12. If this difference is added to the computed conductivity for heartwood specimens, the test values and computed values of K for these four woods will be very close. Only the heartwood specimens of these woods, however, showed this amount of variation. In a few runs on heartwood specimens of oak the computed

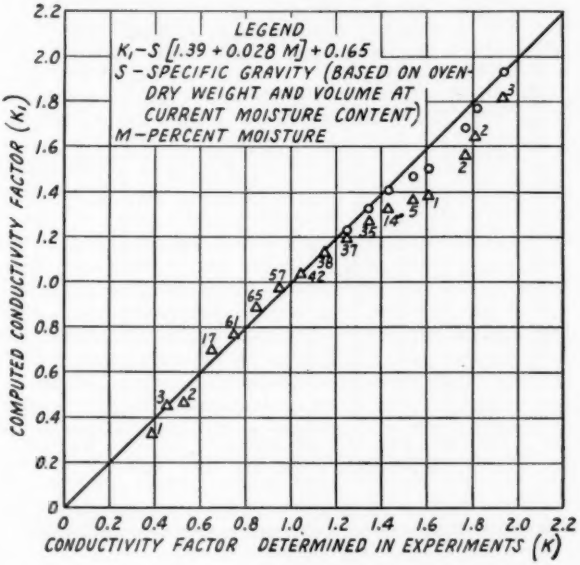

FIG. 5. AVERAGE COMPUTED CONDUCTIVITY PLOTTED AGAINST AVERAGE CONDUCTIVITY DETERMINED IN TESTS ON WOOD WITH DIFFERENT AMOUNTS OF MOISTURE

(Circles show averages when 0.12 is added to computed values (K_1) for heartwood of greenheart, sugar maple and oak. Numbers opposite points show number of runs in average. Data are averaged for each 0.10 difference in experimental conductivity factors.)

conductivity checked very closely with the conductivity found in the tests but more commonly the computed value was a little lower.

The effect of extractives in green oak and maple was apparently obscured by the high moisture content of the specimens since the computed values of K were, in general, very close to those determined in the tests and, as shown in Table 3, varied less than 5 per cent.

Cypress showed a more conspicuous variation in conductivity than many of the other woods tested and it is probable that extractives, such as the oil in the wood, affect the results in this case.

Douglas-fir was more variable than the southern pines and, in general, had

a trend toward a little lower conductivity than other woods of similar specific gravity and moisture content. It was observed that the oven-dry specimens had somewhat lower conductivity for wood of that specific gravity, indicating that there was something inherent in the wood structure affecting the conductivity of this species. The tests on southern pine included a number of runs on resinous wood, but the increase in conductivity was, in most cases, in about the same proportion as the increase in specific gravity because of the

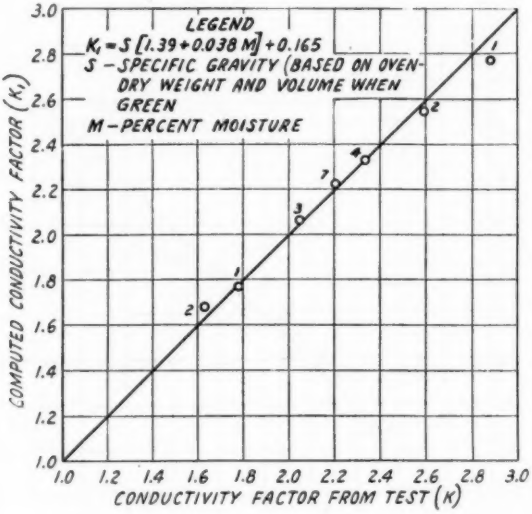

FIG. 6. AVERAGE COMPUTED CONDUCTIVITY PLOTTED AGAINST AVERAGE CONDUCTIVITY DETERMINED IN TESTS ON GREEN WOOD

(Data averaged for each 0.20 difference in experimental conductivity factors. Numbers opposite points show number of runs in average.)

presence of resin. Apparently the conductivity of the resin was not much different from that of the wood substance.

Fig. 5 shows the average computed conductivity plotted against the average conductivity determined in test for specimens having a moisture content ranging from about 2 to 33 per cent. The values are averaged for each 0.1 difference in conductivity. Fig. 6 shows similar data plotted for the tests on green specimens. If the average values computed and the average values determined in the experiments were the same, the plotted points in Figs. 5 and 6 would, of course, fall on the 45 deg line. It may be noted that most of the points in Fig. 5 with the exception of the last few near the upper end of the line, fall on or very close to the 45 deg line. The last points tend to diverge, mainly because of the relatively higher conductivity found in the heartwood of the greenheart, oak and maple specimens. When 0.12 is added

to the computed conductivity for these woods the points fall much closer to the line, as shown by the small circles.

Table 3 shows that the computed and test values for the green material are very nearly the same. In these runs on green wood, the moisture content was 50 per cent or more and the factor 0.038 was used for Z in making the computations.

The data in Table 7 give the variations between the computed conductivity for the different species and the values found in the tests when the moisture content range was from 0 to approximately the fiber-saturation point. The data in this table show that in the total of 84 runs made on oven-dry specimens 57 per cent of the computed conductivity values were within 5 per cent of those determined in the tests; about 29 per cent more were between 5 and 10 per cent; and less than 4 per cent were in the range of maximum variation, which was between 15 and 20 per cent of the values found in the tests.

In the group of 385 runs made on specimens with varying amounts of moisture up to a maximum of about 33 per cent, about 42 per cent of the computed values were within 5 per cent of the test values; 33 per cent more were between 5 and 10 per cent; 20 per cent were between 10 and 15 per cent; and less than 2 per cent were in the range of maximum variation of 20 to 25 per cent. If 0.12 is added to the computed values of conductivity for the heartwood specimens of red and white oak, maple, and greenheart about 50 per cent of the computed conductivity values are within 5 per cent of the values found in the tests; about 31 per cent within 5 and 10 per cent; and 14 per cent between 10 and 15 per cent of the conductivity determined in the tests. Altogether, about 95 per cent of the computed conductivity values varied less than 15 per cent from the values found in the tests on wood having varying amounts of moisture up to the fiber-saturation point. Nearly 80 per cent varied less than 10 per cent from the experimental values.

Fig. 7 shows the relation of conductivity, moisture content, and specific gravity for moisture values ranging from 0 to 30 per cent. The specific gravity and moisture content can be easily determined as previously explained under the sub-heading, *Conditions of Test.* It is often more convenient to determine the original weight of wood per unit volume instead of the specific gravity; the original weight being the weight of the wood and the water it contains at the current moisture content, M. Fig. 8 was therefore prepared to show the relation of weight of wood and conductivity for moisture content intervals ranging from 0 to 30 per cent. The original weight of wood in pounds per cubic foot (actual dimensions) is shown on the bottom scale and the corresponding weight in pounds per square foot and 1 in. in thickness is shown at the top. The approximate average moisture content must of course be known or assumed in order to use this chart. The data for Figs. 7 and 8 were computed by means of Equation (4). Fig. 8 illustrates the point that data showing the original weight per unit volume and conductivity only, are not sufficient since the weight will depend both on specific gravity and moisture content. For this reason either the moisture content or the specific gravity S should also be given in addition to the weight per unit volume. Either Fig. 7 or 8 will be found convenient for finding the approximate conductivity of any particular species, or the conductivity factor can be computed by means of Equation (4) when the moisture content is under 40 per cent.

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TABLE 7—VARIATION OF COMPUTED VALUES OF CONDUCTIVITY FROM VALUES FOUND IN TESTS

(Moisture Content Range 0 to Fiber Saturation Point)

SPECIES	NO. OF RUNS IN WHICH COMPUTED CONDUCTIVITY IS WITHIN THE FOLLOWING PERCENTAGE VARIATION FROM VALUE OF K DETERMINED IN EXPERIMENTS					NO. OF RUNS ON OVEN-DRY WOOD	NO. OF RUNS ON WOOD CONTAINING MOISTURE
	0 to 5 Per Cent	5 + to 10 Per Cent	10 + to 15 Per Cent	15 + to 20 Per Cent	20 + to 25 Per Cent		
Ash, white (Fraxinus americana)							
(Oven-dry wood)	...	...	...	...	...	...	...
(Wood having 12–18 per cent moisture)	6	...	1	...	...	...	7
Aspen, bigtooth (Populus grandidentata)							
(Oven-dry wood)	2	2	1	...	...	5	...
(Wood having 9–17 per cent moisture)	3	3	2	2	...	...	10
Baldcypress (Taxodium distichum)							
(Oven-dry wood)	1	3	...	1	...	5	...
(Wood having 7–15 per cent moisture)	6	12	4	4	4	...	30
Balsa (Ochroma sp.)							
(Oven-dry wood)	1	2	1	...	...	4	...
(Wood having 3–10 per cent moisture)	3	1	...	2	...	...	6
Basswood, American (Tilia glabra)							
(Oven-dry wood)	7	...	...	...	...	7	...
(Wood having 3–12 per cent moisture)	7	9	4	...	...	...	20
Birch, yellow (Betula lutea)							
(Oven-dry wood)	...	...	...	...	...	...	...
(Wood having 6–12 per cent moisture)	6	4	1	...	...	...	11
Douglas-fir (Pseudotsuga taxifolia)							
(Oven-dry wood)	1	4	2	1	...	8	...
(Wood having 7–33 per cent moisture)	14	14	19	4	1	...	52
(Plywood having 4–12 per cent moisture)	5	1	4	2	...	...	12
Elm, rock (Ulmus thomasii)							
(Oven-dry wood)	1	...	...	...	...	1	...
(Wood having 12–32 per cent moisture)	6	3	...	...	...	...	9
Fir, white (Abies sp.)							
(Oven-dry wood)	1	...	1	...	...	2	...

Greenheart[a] (Ocotea rodioei)							
(Oven-dry wood)	...	...	...	...	...	...	...
(Wood having 16–20 per cent moisture)	2	2	...	...	...	...	4
Hemlock, western (Tsuga heterophylla)							
(Oven-dry wood)	1	1	...	...	...	2	...
(Wood having 13–30 per cent moisture)	1	2	1	...	1	...	5
Larch, western (Larix occidentalis)							
(Oven-dry wood)	2	1	...	...	...	3	...
(Wood having 11–13 per cent moisture)	8	4	1	...	...	...	13
Maple, sugar[a] (Acer saccharum)							
(Oven-dry wood)	5	...	...	...	...	5	...
(Wood having 5–28 per cent moisture)	11	9	6	1	...	...	27
Oak, red[a] (Quercus sp.)							
(Oven-dry wood)	1	3	...	1	...	5	...
(Wood having 6–20 per cent moisture)	3	3	6	...	...	...	12
Oak, white[a] (Quercus sp.)							
(Oven-dry wood)	...	...	...	...	...	...	...
(Wood having 5–20 per cent moisture)	1	6	11	...	...	...	18
Pine, southern yellow (Pinus sp.)							
(Oven-dry wood)	5	1	1	...	...	7	...
(Wood having 5–24 per cent moisture)	29	26	3	1	...	...	59
Pine, white (Pinus sp.)							
(Oven-dry wood)	5	3	2	...	...	10	...
(Wood having 4–18 per cent moisture)	8	8	4	...	...	...	20
Prima vera (Tabebuia bonnell-smithii)							
(Oven-dry wood)	1	...	...	...	...	1	...
(Wood having 1–15 per cent moisture)	7	2	...	...	...	...	9

[a] If 0.12 is added to all computed values for the runs on heartwood specimens of maple, oak and greenheart the following results are obtained:

Totals							
(Oven-dry wood)	50	22	10	2	...	84	...
Per cent of total	59.5	26.2	11.9	2.4	...	...	...
(Wood having varying amounts of moisture)	192	118	54	15	6	...	385
Per cent of total	49.9	30.7	14.0	3.9	1.5	...	...

TABLE 7—VARIATION OF COMPUTED VALUES OF CONDUCTIVITY FROM VALUES FOUND IN TESTS—*Concluded*

(*Moisture Content Range 0 to Fiber Saturation Point*)

SPECIES	NO. OF RUNS IN WHICH COMPUTED CONDUCTIVITY IS WITHIN THE FOLLOWING PERCENTAGE VARIATION FROM VALUE OF *K* DETERMINED IN EXPERIMENTS					NO. OF RUNS ON OVEN-DRY WOOD	NO. OF RUNS ON WOOD CONTAINING MOISTURE
	0 to 5 Per Cent	5 + to 10 Per Cent	10 + to 15 Per Cent	15 + to 20 Per Cent	20 + to 25 Per Cent		
Redcedar, western (Thuja plicata)							
(Oven-dry wood)	3	...	...	...	...	3	...
(Wood having 10–20 per cent moisture)	8	3	5	...	...	...	16
Redwood (Sequoia sempervirens)							
(Oven-dry wood)	6	1	1	...	...	8	...
(Wood having 6–13 per cent moisture)	8	7	...	...	...	...	15
Spruce, Engelmann (Picea engelmannii)							
(Oven-dry wood)	3	1	...	...	...	4	...
(Wood having 12–13 per cent moisture)	8	3	1	...	...	...	12
Tanguile (Shorea polysperma)							
(Oven-dry wood)	2	2	...	...	...	4	...
(Wood having 6–15 per cent moisture)	5	1	...	...	...	...	6
Miscellaneous species							
(Oven-dry wood)	...	...	...	...	...	...	...
(Wood having 7–16 per cent moisture)	5	3	4	...	...	...	12
Totals							
(Oven-dry wood)	48	24	9	3	...	84	...
Per cent of total	57.1	28.6	10.7	3.6	...	...	...
(Wood having varying amounts of moisture)	160	126	77	16	6	...	385
Per cent of total	41.5	32.8	20.0	4.2	1.5	...	...

Equation (5) can be used in computing the approximate conductivity of green wood when the moisture content is about 40 per cent or higher.

When wood seasons from the green condition to any given moisture content below the fiber-saturation point, the reduction in moisture will tend to reduce

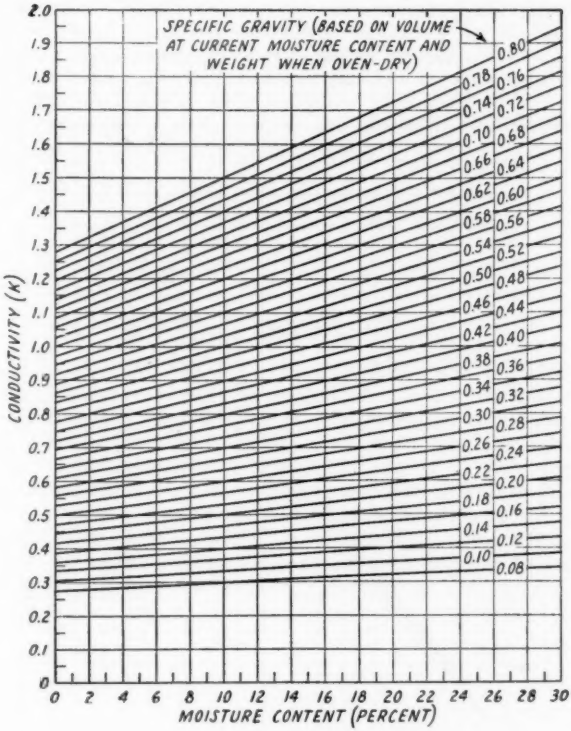

FIG. 7. RELATION BETWEEN COMPUTED CONDUCTIVITY AND MOISTURE CONTENT FOR WOOD HAVING DIFFERENT SPECIFIC GRAVITY VALUES

(Specific gravity based on volume at current moisture content and weight when oven-dry. Conductivity computed from formula $K = S\ [1.39 + 0.028M] + 0.165$.)

the heat conductivity. On the other hand, the specific gravity will increase because of shrinkage in seasoning and this will tend to increase the conductivity. Very often it is desirable to compare the conductivity of one species with that of another when the wood has a particular moisture content, and it is also of interest to know how the conductivity of a given wood changes in going from the green to the oven-dry condition.

Shrinkage studies made at the Forest Products Laboratory show that the

percentage volumetric shrinkage (Vs) that occurs in seasoning from the fiber-saturation point to any moisture content below the fiber-saturation point is approximately proportional to the loss of water from the cell walls. Although some woods having the same specific gravity based on volume when green and weight when oven-dry may not shrink to quite the same extent between the green and oven-dry conditions, for most species the differences in shrinkage

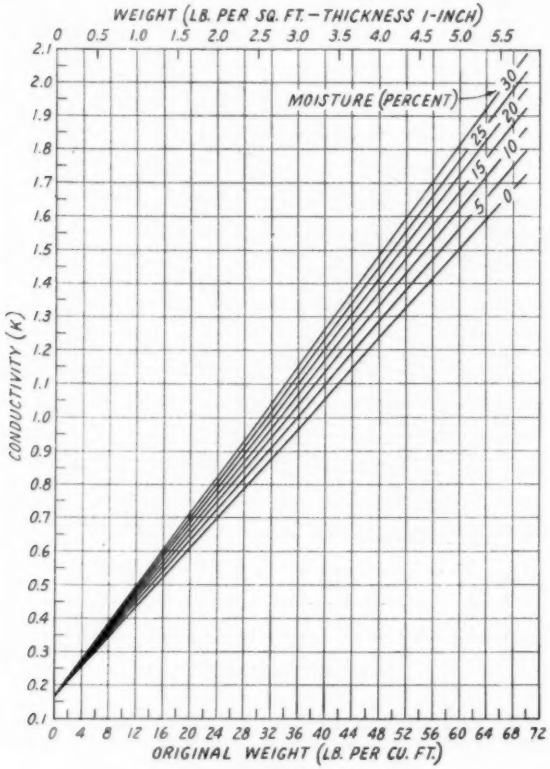

FIG. 8. RELATION BETWEEN COMPUTED CONDUCTIVITY AND ORIGINAL WEIGHT OF WOOD (WEIGHT AT CURRENT MOISTURE CONTENT M) WITH MOISTURE CONTENT RANGING FROM 0 TO 30 PER CENT

Conductivity computed from formula, $K = S\ (1.39 + 0.028M) + 0.165$

are usually small when the specific gravity of the green wood is the same. It is therefore sufficiently accurate for practical purposes to compute the shrinkage and the corresponding specific gravity on the basis that shrinkage is proportional to the loss of water from the cell walls.

The approximate volumetric shrinkage would then be computed from:

$$Vs = (Sg)\left[\left(\frac{1}{1.115}\right)(30 - M)\right]$$

$$\text{or since } \frac{1}{1.115} = 0.9, \text{ approximately } Vs = Sg\,[0.9\,(30 - M)]$$

In this expression Vs is the percentage volumetric shrinkage; Sg is the specific gravity based on volume when green and weight when oven-dry; 1.115 is the density of the water in the fiber walls when they are saturated and M is the percentage moisture to which the wood is seasoned.

If Sm represents the specific gravity after seasoning to a moisture content M the value of Sm is found from the relation

$$Sm = \left[\frac{Sg}{1 - \frac{Vs}{100}}\right] = \frac{Sg}{1 - Sg\,[0.009\,(30 - M)]}$$

Likewise Sg can be computed if Sm is known or assumed since

$$Sg = \frac{Sm}{1 + (Sm)\,[(0.009)\,(30 - M)]}$$

Average values of Sg for a considerable number of the more important commercial native species are given in Table 8. Similar data for other woods are given in U.S.D.A., Tech. Bul. 479.[10] Since seasoned wood soaked in water until it is thoroughly wet will swell to about green dimensions, the value of Sg can be readily determined for samples from any timber regardless of the degree of seasoning, by assuming the green volume to be the same as the volume of the sample after soaking to maximum swelling in water.

If the computed specific gravity Sm is substituted for S in the equation $K = \mathrm{S}\,(1.39 + 0.028M) + 0.165$ the approximate conductivity K can be computed for wood at any given moisture content using the specific gravity Sg based on the green wood. Substituting for Sm its equivalent

$$\left[\frac{Sg}{1 - \left(\frac{Vs}{100}\right)}\right]$$

the equation for conductivity becomes:

$$K = \frac{Sg\,[1.39 + 0.028M]}{1 - Sg\,[0.009(30 - M)]} + 0.165$$

This equation was used for computing the curves in Fig. 9 which show the relation between the computed conductivity and moisture content for woods having various specific gravity values based on volume when green and weight when oven-dry. The following example illustrates the use of Fig. 9. Assume it is desired to find approximately the average conductivity of loblolly pine at 15 per cent moisture. Table 8 shows the average specific gravity of this wood when green is 0.47. By following the vertical line for 15 per cent moisture to a point halfway between the lines for specific gravities of 0.46 and 0.48 the conductivity opposite this point (read on left hand scale) is shown to be about 1.07. If we wish to compare the conductivity of this species

TABLE 8—AVERAGE SPECIFIC GRAVITY OF VARIOUS SPECIES OF SOFTWOODS AND HARDWOODS

SPECIES	SPECIFIC GRAVITY BASED ON VOLUME WHEN GREEN AND WEIGHT WHEN OVEN DRY = *Sg*
HARDWOODS	
Ash, white (Fraxinus americana)	0.55
Basswood, American (Tilia glabra)	0.32
Beech, American (Fagus grandifolia)	0.56
Birch, yellow (Betula lutea)	0.55
Blackgum (Nyssa sylvatica)	0.46
Chestnut, American (Castanea dentata)	0.40
Elm, American (Ulmus americana)	0.46
Elm, rock (Ulmus thomasii)	0.57
Hackberry (Celtis occidentalis)	0.49
Hickory, mockernut (Hicoria alba)	0.64
Maple, silver (Alcer saccharinum)	0.44
Maple, sugar (Acer saccharum)	0.56
Oak, red (commercial) (Quercus sp.)	0.56
Oak, white (commercial) (Quercus sp.)	0.59
Pecan (Hicoria pecan)	0.60
Sweetgum (Liquidambar styraciflua)	0.44
Sycamore, American	0.46
Tupelo, water (Nyssa aquatica)	0.46
Walnut, black (Juglans nigra)	0.51
Yellowpoplar (Liriodendron tulipifera)	0.38
SOFTWOODS	
Baldcypress (Taxodium distichum)	0.42
Douglas-fir, coast (Pseudotsuga taxifolia)	0.45
Fir, white (Abies sp.)	0.37
Hemlock, eastern (Tsuga canadensis)	0.38
Hemlock, western (Tsuga heterophylla)	0.38
Larch, western (Larix occidentalis)	0.48
Pine, eastern white (Pinus strobus)	0.34
Pine, jack (Pinus banksiana)	0.39
Pine, loblolly (Pinus taeda)	0.47
Pine, lodgepole (Pinus contorta)	0.38
Pine, longleaf (Pinus palustris)	0.54
Pine, ponderosa (Pinus ponderosa)	0.38
Pine, red (Pinus resinosa)	0.44
Pine, shortleaf (Pinus echinata)	0.46
Pine, slash (Pinus caribaea)	0.56
Pine, sugar (Pinus lambertiana)	0.35
Pine, western white (Pinus monticola)	0.36
Redcedar, eastern (Juniperus virginiana)	0.44
Redcedar, western (Thuja plicata)	0.31
Redwood (Sequoia sempervirens)	0.38
Spruce, Engelmann (Picea engelmannii)	0.31
Spruce, Sitka (Picea sitchensis)	0.37
Spruce, white (Picea glauca)	0.37
Tamarack (Larix laricina)	0.49
White-cedar, northern (Thuja occidentalis)	0.29

with that of another wood, for example, yellow birch, Table 8 shows the average specific gravity (based on volume when green) of the latter species is

0.55. Fig. 9 shows that the conductivity of yellow birch at 15 per cent moisture would be about 1.24 or nearly 16 per cent more than the conductivity of loblolly pine at this moisture content. It should be noted that the specific gravity values *Sg* shown in Fig. 9 are based on oven-dry weight and volume when green while the specific gravity values given in Fig. 7 correspond to

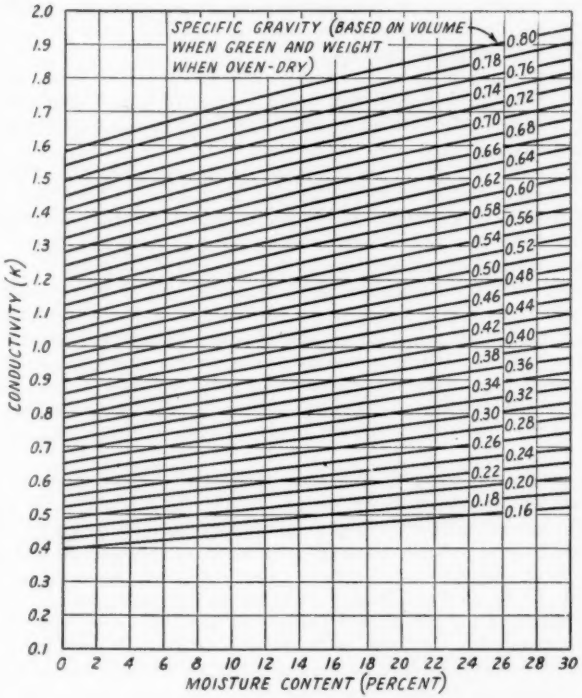

FIG. 9. RELATION BETWEEN COMPUTED CONDUCTIVITY AND MOISTURE CONTENT FOR WOODS HAVING VARIOUS SPECIFIC GRAVITY VALUES BASED ON VOLUME WHEN GREEN AND WEIGHT WHEN OVEN-DRY

Sm and are based on oven-dry weight and volume at current moisture content. The latter figure is convenient to use when the specific gravity and moisture content of seasoned wood is determined or assumed.

It may be noted from Fig. 9 that loss in moisture content reduces the conductivity to a greater extent than the accompanying change to higher density increases it, hence the net result is a decrease in conductivity as wood seasons from the green to the oven-dry condition.

The small number of tests made on Douglas-fir specimens to compare conductivity in the radial and tangential directions, showed no pronounced differ-

ences although, in practically all cases, there appeared to be a tendency for higher conductivity in the tangential direction. For average wood, however, there would probably be no important difference in conductivity for these two directions. On the other hand there is a very marked difference in the conductivity in a longitudinal direction compared with that at right angles to the fibers. In several runs on Douglas-fir and red oak specimens in which the moisture content ranged from about 6 to 15 per cent, the conductivity in the longitudinal direction was from about 2¼ to 2¾ times the conductivity in the radial or tangential direction.

Experiments made in special runs to study the effect of knots, checks, and cross-grain structure indicated that small knots, when not numerous, had no important influence on conductivity but large knots had a tendency to increase the conductivity. It was also noted that there was little if any effect on conductivity when the specimens had small checks, such as often occur in wood while seasoning. Wood with pronounced cross-grain (from face to face) showed increased conductivity, as would be expected. In wood having such cross-grain, there is a longitudinal component that increases the conductivity to a greater or less extent depending on the angle of cross-grain.

Results of the experiments on specimens of Douglas-fir plywood indicate that the thin film of glue between the wood surfaces has no important effect on conductivity. This might be expected because of the very slight thickness of the glue coating. Even a marked difference in the conductivity of the glue in comparison with the conductivity of the wood would make little difference in the conductivity through the entire thickness because the total thickness of the glue is insignificant in comparison with the total thickness of the wood.

A few runs were made to investigate the effect of differences in average temperature on the conductivity of the wood. The temperature differences between the hot and cold sides ranged from about 22 to 96 F and the differences in the average temperatures obtained in runs on a given pair of test specimens (differences in average of hot-plate plus cold-plate temperatures) varied from about 12 to 25 F. Changes in temperature were obtained by varying the temperature of the hot plate. These runs were made on both air-seasoned and oven-dry woods. In most cases, there was a small increase in conductivity with increase in average temperature, but the increase varied from nearly zero to a maximum of less than 4 per cent. In other words, for the range of hot- and cold-plate temperatures used, there was but little change in conductivity when the difference in temperature between the hot- and cold-plate sides was more than doubled by increasing the hot-plate temperature. This indicates that differences in the average temperatures of wood which are normally encountered, have no important bearing on heat conductivity.

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No. 1185

EFFECT OF INSULATION ON PLANT PERFORMANCE IN THE RESEARCH RESIDENCE

By A. P. KRATZ * AND S. KONZO,** URBANA, ILLINOIS

INTRODUCTION

THE investigations in gravity warm-air heating and forced warm-air heating in the Research Residence at the University of Illinois prior to 1939 were made in an uninsulated structure. During the summer of 1939 the exposed side walls and ceilings exposed to unheated space were fully insulated with mineral wool.

For the purpose of direct comparison between the results obtained with the uninsulated and insulated structure, the same forced-air heating plant, with no changes in the duct system, the baseboard warm air registers, the centrifugal fan, the stoker-fired warm air furnace, or the automatic control system was used in both cases.

The tests made during the heating season of 1939-1940 were confined to studies of the performance and operating characteristics of the existing heating plant when the heat loss from the structure was materially reduced by the application of insulation and storm sash. Since the existing heating plant was adequate in capacity to heat the uninsulated structure, the use of the same plant in connection with the reduced heat loss was equivalent to the use of an oversized heating plant in the insulated structure.

DESCRIPTION OF RESEARCH RESIDENCE AND HEATING EQUIPMENT

The Research Residence, shown in Fig. 1, and the forced-air heating plant have been completely described in a previous publication.[1] The Residence is a three-story structure of standard frame construction. The total space heated, including a sun room, amounted to 17,540 cu ft.

In insulating the sidewalls and ceiling, wherever feasible, mineral wool batts were laid between the studding or joists. Wherever batts could not be

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[1] University of Illinois, *Engineering Experiment Station Bulletin No. 266* by A. P. Kratz and S. Konzo, 1934.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, San Francisco, Calif., June, 1941.

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conveniently used, nodulated mineral wool, having a density of about 5 lb per cubic foot, was blown into place. Owing to the fact that the house was constructed with 6-in. studs, the insulation in the walls and ceilings was 5⅝ in. thick. The insulation in the knee-walls between heated and unheated spaces was 3⅝ in. thick. On account of the presence of ductwork in some of the outside studding spaces, 95 sq ft of exposed wall remained uninsulated as compared with 2,164 sq ft that were insulated. The over-all coefficient of heat transfer for the exposed wall was 0.047 Btu per square foot per hour

FIG. 1. WARM AIR HEATING RESEARCH RESIDENCE, URBANA, ILL.

per degree Fahrenheit and that for the ceilings was 0.048 Btu per square foot per hour per degree Fahrenheit.

The double-hung windows were not equipped with weather strips, and during the tests they remained tightly locked. One series of tests was run with the Residence equipped with storm windows and door. All of the 50 windows on the three stories of the Residence, with the exception of two small quarter-round windows in the east dormitory, were provided with tightly fitting storm sash. Felt stripping was placed along all four contact edges and the storm sash was drawn up tightly to the window casing. The front entrance was equipped with a storm door.

The heating plant consisted of a cast-iron, circular-radiator warm-air furnace used in connection with a forced-air heating system. The principal

dimensions of the furnace are shown in Table 1. A centrifugal fan delivered 1675 cu ft of air per minute through the furnace casing and duct system.

The furnace was fired by means of a coal stoker of the underfeed type, which was inserted through the ash pit door. A balanced check damper was installed in the clean out of the chimney. The fuel burned was 1 in. x 10 mesh stoker coal, washed and oil treated and was obtained from Saline County, Ill. The feed rate of the stoker was about 26 lb per hour, and the air input was sufficient to provide a burning rate of approximately 13 lb per hour during *on-periods*.

The heating plant was controlled by means of a room thermostat operating to start and stop the stoker motor, and to start and stop the circulating fan as shown in Fig. 2. This room thermostat was of the heat-anticipating type, and was used in conjunction with two bonnet thermostats which served as high

TABLE 1—DIMENSIONS AND AREAS OF FURNACE USED

Grate diameter	23 in.
Grate area	2.88 sq ft
Firepot diameter	27 in.
Heating surface	
Ash-pit	1.36 sq ft
Firepot	8.10 sq ft
Dome	18.87 sq ft
Radiator	31.93 sq ft
Total	60.26 sq ft
Ratio of heating surface to grate area	20.9
Casing diameter	50 in.
Free area through casing	4.97 sq ft
Combustion space[a]	8.8 cu ft
Free area through over-fire damper	4.9 sq in.

[a] Combustion space is defined in this case as the space above the hearth level, including the dome, but not the feed neck.

and low limit controls for the temperature of the air in the furnace bonnet. When the room thermostat operated to start the fan and burner, one bonnet thermostat prevented the fan from starting unless the bonnet temperature was above 125 F. The other bonnet thermostat prevented the burner from starting unless the bonnet temperature was below 150 F.

METHOD OF CONDUCTING TESTS

The average of the air temperatures in all of the rooms of the house was maintained at 72 F both day and night. Observations of weather, indoor room air temperatures, room relative humidities, and other incidental data were made daily at 7:00 a.m., 11:00 a.m., 4:00 p.m., and 10:00 p.m. Complete data were obtained for each 24-hour test period on the fuel consumption, weight of ash and clinkers removed, the total integrated time of operation of the fan and of the stoker, the total electrical input to the fan and burner motors, and the total number of *on-periods* of both the circulating fan and the stoker. Daily observations were made of the volume of air circulated, and the filters were cleaned with sufficient frequency to maintain the air volume

constant. In addition, continuous records of temperatures, CO_2, and the index of smoke density were obtained for each 24-hr period. Each day at 11 a.m., the clinkers were removed, the fuel bed was levelled, and the hopper was filled with coal. During extremely mild weather no attention was given the fuel bed or hopper, except as required every two or three days. By means of the balanced check damper, the draft in the smoke pipe was maintained at approximately 0.05 in. of water. For each series of tests, data were obtained over a wide range of outdoor weather conditions.

The following series of tests were run: *Series 4-38,* which was discussed in a previous paper,[2] and for which the Residence was not insulated and the

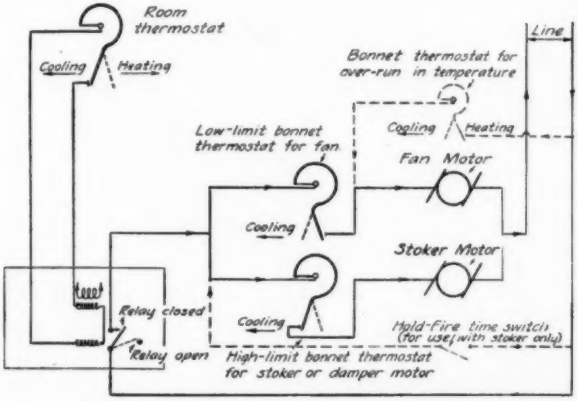

FIG. 2. SIMPLIFIED WIRING DIAGRAM FOR CONTROL OF FAN MOTOR AND STOKER MOTOR
(*Research Residence Installation, Seasons of 1937-40*)

windows were not equipped with storm sash; *Series 1-39,* for which the Residence was insulated, but the windows were not equipped with storm sash; and *Series 2-39,* for which the Residence was insulated and the windows were equipped with storm sash. In no case were any changes made in the quantity of air circulated, in the feed rate and the rate of air supply to the stoker, in the settings of the automatic controls, or in the method of distribution of air from the baseboard registers.

RESULTS OF TESTS

Calculated Values of Heat Loss: The calculated values of the heat losses from the Residence, based on coefficients computed from the conductivities given in THE ASHVE GUIDE and on a temperature difference between indoors and outdoors of 80 F, are given in Fig. 3. The heat losses for each of the three stories of the Residence, as well as those for the entire struc-

[2] Performance of a Stoker-Fired Warm-Air Furnace as Affected by Burning Rate and Feed Rate, by A. P. Kratz and S. Konzo. (ASHVE TRANSACTIONS, Vol. 46, 1940.)

ture, are shown in graphical form. As an example, in *Case A*, for which the house was not insulated and storm sash were not used, the calculated heat losses were 32,385 Btu per hour for the third story, 49,780 for the second story, 67,675 for the first story, and 149,840 for the entire structure. In the usual two story structure of compact design and without exposed wings, the heat loss from the second story rooms is generally greater than that from the first story rooms. In the case of the Research Residence, however, the exposed sun room added a material heat loss to that from the remainder

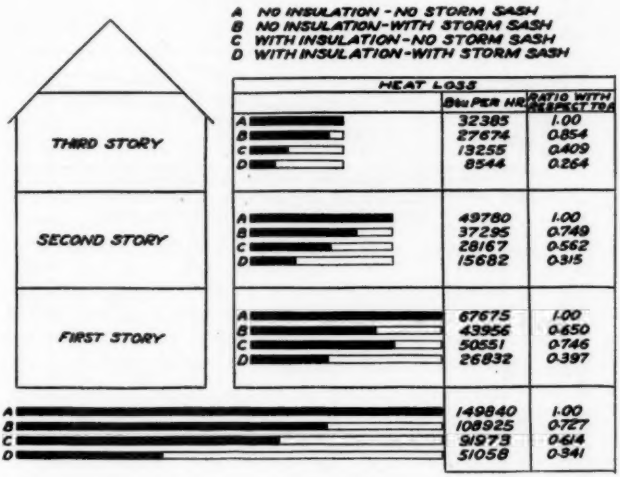

FIG. 3. CALCULATED HEAT LOSSES FROM RESEARCH RESIDENCE FOR 80 F DIFFERENCE

of the rooms on the first story, with the result that the total heat loss on the first story was much greater than that on the upper stories.

In *Case C*, for which the house was insulated and storm sash were not used, the calculated heat losses were 13,255 Btu per hour for the third story, 28,167 for the second story, 50,551 for the first story, and 91,973 for the entire structure. The ratios of each of these values to the corresponding values obtained for *Case A* are given in the last column in Fig. 3, and were 0.409, 0.562, 0.746, and 0.614 for the third story, second story, first story and entire structure respectively. That is, the estimated percentage reduction in the heat loss effected by the application of insulation to the sidewalls and ceiling was considerably greater on the third and second stories than it was on the first story. Furthermore, while heat losses from the second story of the more common type of two story structure of compact design are greater than those from the first story, usually the application of insulation to sidewalls and ceiling also results in a proportionally greater reduction on the second story than on the first story.

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After the Residence had been insulated, and before any changes had been made in the settings of the dampers in the branch ducts leading to the twelve warm air registers, a few preliminary tests were made to determine whether the installation of the insulation had resulted in any material lack of uniformity in the room temperatures. From the reductions in heat loss shown

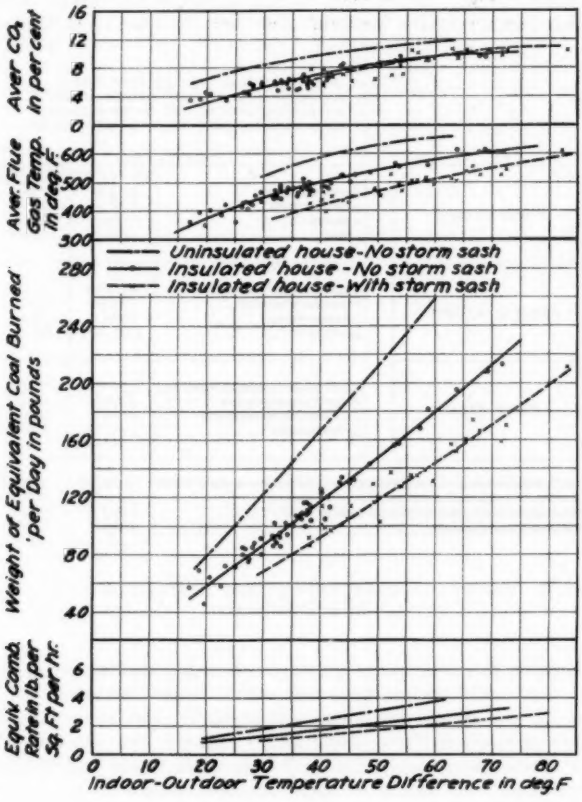
Aver. CO2 in per cent
Aver. Flue Gas Temp. in deg. F.
Weight of Equivalent Coal Burned per Day in pounds
Equiv. Comb. Rate in lb. per Sq. Ft. per hr.
Uninsulated house-No storm sash
Insulated house-No storm sash
Insulated house-With storm sash
Indoor-Outdoor Temperature Difference in deg. F

FIG. 4. FURNACE PERFORMANCE

in *Case C* in Fig. 3, it would be reasonable to expect that after the application of insulation, the room temperatures in the third and second story rooms would be higher than those in the first story rooms. Measurements of the air temperatures in the various rooms of the Residence confirmed this hypothesis by proving that the temperatures in the third story rooms were from 5 to 7 F higher, and the temperatures in the second story rooms were from 1 to 3 F higher, than those in the first story rooms.

Hence, in the case of many existing homes in which the heating facilities in upper story rooms are inadequate, the application of insulation to sidewalls and ceiling should not only reduce the total heat losses, but should also tend to bring about a better balance in temperatures in the rooms on the two stories.

A comparison of *Case B* in Fig. 3 with *Case A* indicates that the application of storm sash alone on an uninsulated house is relatively more effective in reducing the heat losses from the first story rooms than it is on the upper stories. Whereas, a comparison of *Case D* with *Case A* indicates that the combined application of insulation and storm sash would reduce the heat losses approximately in proportion, on each of the three stories. In any case, after the application of storm sash or insulation to an existing structure, some amount of readjustment of the heat inputs to the various rooms will be necessary.

After the preliminary observations were made, the dampers in the ducts of the forced-air heating system in the Research Residence were readjusted to reduce the amount of warm air delivered to the second and third story rooms. The damper adjustments were easily made and no difficulty was experienced in obtaining a uniform air temperature of about 72 F in all of the rooms in the Residence.

Fuel Consumption: The calculated values of the total heat loss for *Cases A* and *C*, given in Fig. 3, indicate an estimated possible reduction in heat loss effected by insulating the sidewalls and ceiling amounting to 38.6 per cent. The data presented in Fig. 4, showing the weight of equivalent coal (13040 Btu per lb) burned per day, were obtained from actual tests and afforded a means of comparing the actual reduction in fuel consumption with the calculated reduction in heat loss. The average reduction in fuel consumption obtained over a wide range of weather conditions was about 30 per cent, or approximately 0.78 of the estimated reduction in heat loss. The agreement between the actual reduction in fuel consumption and the estimated reduction of heat loss may be considered as close as can be expected considering the uncertainties in many of the assumptions inherent in the calculation of heat losses.

A similar comparison from Fig. 3 of the calculated heat losses for *Case A* and *Case D*, for the latter of which both insulation and storm sash were used, indicates an estimated possible total reduction of 65.9 per cent, or an additional estimated reduction of 27.3 per cent effected by the storm sash alone. The actual reduction in fuel consumption resulting from the use of both insulation and storm sash as shown in Fig. 4 was about 45 per cent, or only 0.68 of the estimated possible reduction in heat loss. The reduction in fuel consumption resulting from the use of storm ash alone was about 15 per cent or only 0.55 of the estimated reduction in heat loss. In this case, the discrepancy between the estimated reduction in heat loss and the actual reduction in fuel consumption was quite large and may be due to the fact that certain factors affected by the actual conditions of operation deviated from those assumed for the calculation of the heat loss.

Part of the discrepancy may be attributed to the uncertainties which necessarily accompany the computation of infiltration losses, a complete discussion of which has been presented in a previous paper.[3] Some of the discrepancy may have resulted from the use of an over-sized stoker-fired furnace in the

[3] Fuel Saving Resulting from the Use of Storm Windows and Doors, by A. P. Kratz and S. Konzo. (ASHVE TRANSACTIONS, Vol. 42, 1936.)

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insulated Residence. The stoker and furnace, which were adequate in size for the uninsulated house, were operated intermittently, by means of the thermostatic control system, to maintain the desired room temperatures in the insulated house. For a given outdoor temperature, the total operating time of the stoker was not as long, as shown in Fig. 5, and the average CO_2 in the flue gas was not as high, as shown in the top of Fig. 4, in the case of the insulated house as they were in the case of the uninsulated house. It is

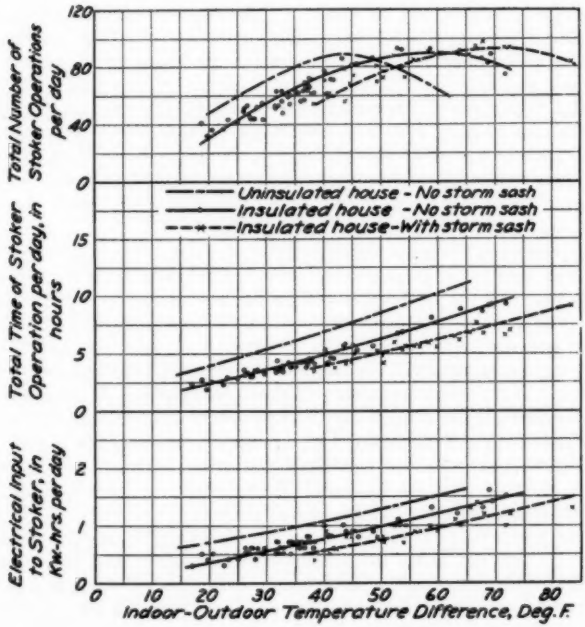

FIG. 5. OPERATING CHARACTERISTICS OF STOKER

possible that in spite of the decrease in the accompanying average flue gas temperatures, as shown in Fig. 4, the over-all combustion efficiency for the stoker, during both the *on-periods* and *off-periods*, was less with the stoker and furnace actually used than it would have been if the equipment had been properly sized for the insulated house. No tests have as yet been made to determine whether the use of a smaller sized stoker and furnace in the insulated Residence would improve the over-all combustion efficiency and thereby reduce the discrepancy between the estimated reduction in heat loss and the actual reduction in fuel consumption. However, previous tests [4] would indicate that if the stoker used had been regulated in the case of the insulated house to

[4] Loc. Cit. Note 2.

burn coal at a rate less than the 13 lb per hour actually employed, some gain in fuel economy would have been attained. In general, if maximum fuel economies are to be obtained after the heat losses from a given house have been reduced by insulation or otherwise, some attention should be given to the matter of insuring the use of a burning rate commensurate with the reduced heat losses. On the whole the results obtained are representative of those arising from the usual practice of insulating the house without changing the heating plant.

Furnace Capacity: As shown by Fig. 4, the average flue gas temperatures for a given day were greatest when the house was not insulated and storm sash were not used, and were the least when the house was insulated and equipped with storm sash. Hence, the application of insulation and storm sash to a house already equipped with a furnace plant is equivalent to lightening the maximum load on the plant, and thereby increasing the life of the furnace equipment.

The furnace and stoker in the Residence had sufficient capacity to satisfy the heating demands of the uninsulated house when the outdoor temperature attained a value of −10 F, corresponding to the design temperature used in Urbana, Ill. By burning the same total amount of coal as that required to heat the uninsulated house at −10 F, the furnace and stoker in the insulated house would have had adequate capacity to satisfy the heating demands for an outdoor temperature of approximately −44 F, which is far in excess of the lowest temperature recorded in Urbana.

In an old house this excess capacity represents a reserve sufficient to provide for the lowest possible extreme of outdoor temperatures in a given locality, while in a new house it represents the margin by which the initial cost of a new plant may be reduced.

Stoker and Fan Operation: The influence of the controls on the fan and stoker motors was such that, irrespective of the size of the furnace relative to the over-all heating demand of the house, the heat output was proportioned approximately to the heat losses occurring in either the insulated or the uninsulated house at a given indoor-outdoor temperature difference. In the case of the insulated, as compared with the uninsulated house, since the heat demand was less, the stoker remained in operation for a shorter period. Hence, at the end of this period the combustion was not as active and the amount of incandescent coke in the fuel bed was less. As a result, the combustion during the *off-period* was also less active, and the temperature in the bonnet at the end of the *off-period* and hence at the start of the subsequent *on-period* was not as high. During the *on-period,* while air was being circulated by the fan, the temperature in the bonnet decreased, and at the end of the *on-period* it was lower than that obtained in the case of the uninsulated house. Consequently, as shown in Fig. 6, the furnace operated at a lower range of bonnet temperatures in the case of the insulated house than it did in the case of the uninsulated house. This lower temperature range, in conjunction with the reduced time of operation, served to proportion the furnace output to the heat loss from the house. As shown by Figs. 5 and 6, the total time of operation of the stoker and fan per day, and the electrical inputs to the stoker motor and fan motor per day were each decreased approximately 25 to 30 per cent after the house was insulated, as compared with the results obtained in the uninsulated house.

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Factors Affecting Comfort: As discussed in a previous publication,[5] the maintenance of a small temperature differential from breathing level to floor is a necessary requirement for comfort. Furthermore, short *off-periods* of the fan are more desirable from the standpoint of comfort than long *off-periods*,

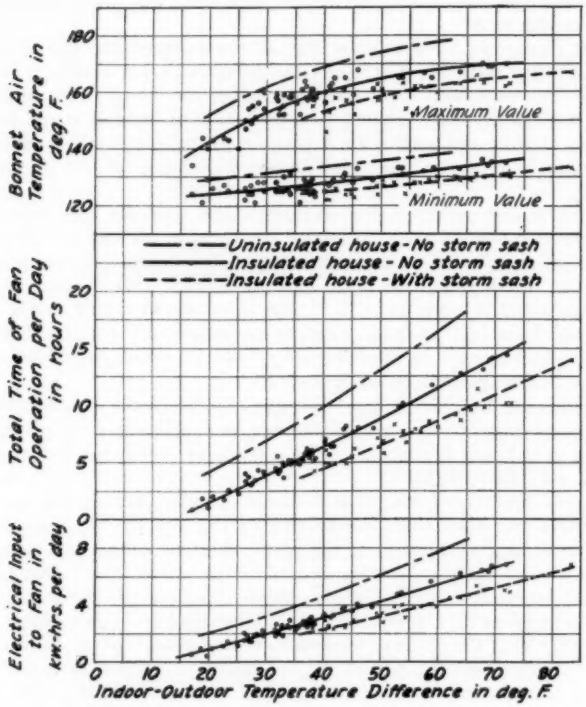

FIG. 6. OPERATING CHARACTERISTICS OF FAN

in that the shorter *off-periods* tend to minimize the temperature variations in the room brought about by cyclic operation. This is particularly true in cold weather, or for temperature differences from indoor to outdoor exceeding approximately 50 F.

The tests made in the uninsulated and insulated Residence afforded a means of comparing the length of *off-periods*, and also the temperature differentials

[5] Study of Methods of Control and Types of Registers as Affecting Temperature Variations in the Research Residence, by A. P. Kratz and S. Konzo. (ASHVE TRANSACTIONS, Vol. 44, 1938.) Investigation of Oil-Fired Forced-Air Furnace Systems in the Research Residence, by A. P. Kratz and S. Konzo. (University of Illinois, *Engineering Experiment Station Bulletin* 318, Nov. 1939.)

as affected by changes in the nature of the structure when no changes were made in the heating plant itself, or in its method of operation. As shown in Fig. 7, for a given temperature difference from indoors to outdoors, the use of insulation brought about an increase in the length of time that the fan

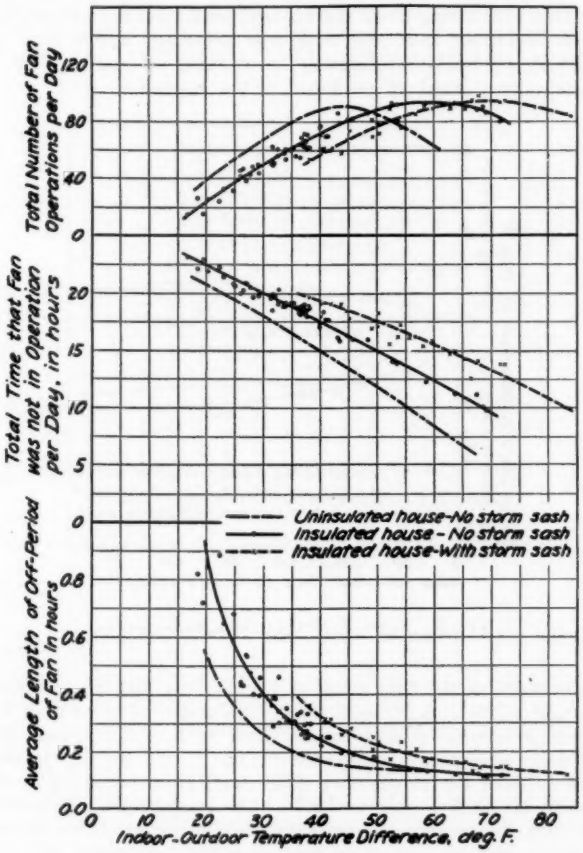

FIG. 7. OFF-PERIODS OF FAN

was not in operation per day. Fortunately, for temperature differences exceeding about 50 F, an increase in the number of fan operations per day was also obtained, with the net result that the average length of the *off-periods* was only slightly increased.

During periods of cold weather, as shown by Fig. 8, the temperature differ-

entials from breathing level and floor were diminished slightly after the house was insulated, probably as a result of the fact that any down currents of air over the exposed walls were not as cold in the case of the insulated as they were in that of the uninsulated house. Hence, any slight tendency for the existence of a cold floor resulting from the increase in the length of the *off-period* of the fan evidently was more than offset by the beneficial effect of the warmer air coming down the exposed walls, and the temperature differential from breathing level and floor was decreased rather than increased. On the whole, no marked differences in comfort, as measured by room air temperatures alone, could be attributed to the insulation.

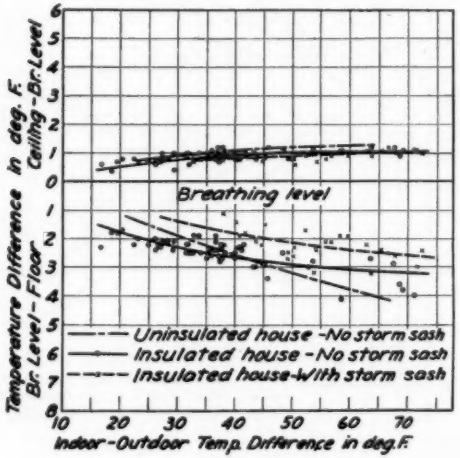
Temperature Difference in deg. F.
Br. Level - Floor
Ceiling - Br. Level
Breathing level
Uninsulated house - No storm sash
Insulated house - No storm sash
Insulated house - With storm sash
Indoor - Outdoor Temp. Difference in deg. F.

FIG. 8. ROOM TEMPERATURE DIFFERENTIALS

In this connection, the data given in Fig. 8 show that the combined use of insulation and storm sash brought about a reduction of more than 1 F in the temperature differential from breathing level to floor. For example, for an indoor-outdoor temperature difference of 60 F, the use of insulation and storm sash reduced the temperature differential from breathing level to floor from a value of 3.8 F to one of 2.3 F. The temperature differentials shown in Fig. 8 were the averages of readings made in eleven different locations in the Residence, and at most were much smaller than those that have been obtained with other arrangements of the heating plant in the same structure. It is possible that in structures which are not as satisfactorily heated as the Residence originally was, and which show much larger temperature differentials from breathing level to floor, the use of insulation and storm sash might show to much better advantage.

The most marked improvement in comfort with the use of insulation was that resulting from the large increase in the inside surface temperatures of the exposed walls. The difference in temperature between the indoor air and the

inside wall surface was plotted, as shown in Fig. 9, against the temperature difference between the indoor air and outdoor air. The temperature differences between indoor air and inside wall surface obtained from test observations on both the uninsulated and insulated wall section are designated by solid lines, and calculated values of the same quantities are designated by broken lines. For both the uninsulated and insulated wall sections, the calculated temperature differences were slightly smaller than the observed differences. The calculated and observed differences are in reasonably good agreement considering the fact that the actual coefficients of heat transmission for

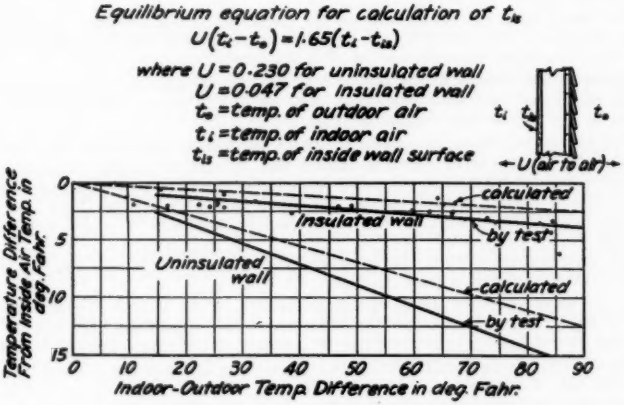
Equilibrium equation for calculation of t_{is}
$U(t_i - t_o) = 1.65(t_i - t_{is})$
where U = 0.230 for uninsulated wall
U = 0.047 for insulated wall
t_o = temp. of outdoor air
t_i = temp. of indoor air
t_{is} = temp. of inside wall surface
t_i t_{is} t_o
← U (air to air) →
Temperature Difference From Inside Air Temp. in deg. Fahr.
0
5
10
15
Insulated wall
calculated
by test
Uninsulated wall
calculated
by test
0 10 20 30 40 50 60 70 80 90
Indoor-Outdoor Temp. Difference in deg. Fahr.

FIG. 9. INSIDE SURFACE TEMPERATURES OF WALL (NORTH WALL, LIVING ROOM)

the wall and the surface coefficient may differ somewhat from those assigned in the calculations.

Test values in Fig. 9 indicate that for an indoor-outdoor temperature difference of 80 F the wall surface temperature was 14.5 F lower than that of the indoor air in the case of the uninsulated wall and only 3.5 F lower in the case of the insulated wall, or an improvement of 11.0 F effected by the insulation. With the large wall areas involved, this warmer wall surface temperature would prove to be a significant factor in reducing the radiation loss from the body, and in increasing the comfort of the occupant, even though the room air temperature was the same. In this connection it is of interest to note that the increase in inside surface temperature of the glass in the windows brought about by the use of storm sash is of even greater magnitude than that shown in Fig. 9 for the wall. The reductions in fuel consumption, shown in Fig. 4, effected by the use of insulation and storm sash would probably have been greater if the tests had been made so that in both cases equal comfort had been maintained in the Residence, rather than equal dry-bulb temperatures. Due to difficulties in evaluating comfort, and to the fact that wall surface temperatures were not measured in all of the

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rooms of the Residence, no such tests were attempted. In any case the fuel economies reported in this paper may be considered as minimum rather than maximum values.

SUMMARY

The following is a summary of the results obtained in the Research Residence and is applicable to the conditions under which the tests were conducted:

1. The installation of insulation resulted in an average saving of approximately 30 per cent in the actual fuel consumption, as compared with an estimated reduction in heat loss of 38.6 per cent.

2. The installation of storm sash on the insulated house resulted in an additional average saving of approximately 15 per cent in the actual fuel consumption as compared with an estimated reduction of 27.3 per cent.

3. The electrical inputs to the stoker and fan motors per day were each decreased from 25 to 30 per cent after the house was insulated, as compared with the results obtained in the uninsulated house.

4. The estimated reductions in heat losses indicated that a greater benefit was obtained from the insulation of the second and third stories than from that of the first. This was confirmed by the fact that temperatures on the second story were from 1 to 3 F higher, and those on the third story were from 5 to 7 F higher, after the installation of insulation and before final adjustment of the dampers, than they were in the uninsulated house.

5. The temperature differential from heating level to floor was only slightly decreased by the application of insulation.

6. At an indoor-outdoor temperature difference of 80 F, the temperature of the inside surface of the exposed walls was increased approximately 11.0 F by the application of insulation, thus resulting in a material potential increase in comfort.

7. The application of insulation without changes in the heating system was equivalent to the use of an oversized furnace. In spite of this, on the whole no difficulties were encountered from overheating or from lack of uniform temperature distribution in the rooms and the results may be considered as representative of those arising from the usual practice of insulating the house without changing the heating plant.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with the investigation of warm-air furnaces and heating systems in the Research Residence [6] at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. This investigation is a cooperative project sponsored jointly by the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is made to D. W. Thomson, formerly Research Graduate Assistant, for his services in conducting the tests and in the reduction of the test data. Acknowledgment is also made to the *National Mineral Wool Association* for their interest and cooperation in insulating the Residence, and to the various manufacturers who cooperated by furnishing instruments and equipment.

[6] The Research Residence in Urbana, Illinois, was built, furnished, and completely equipped specifically for research work in warm-air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

DISCUSSION

O. W. JOHNSON: I would like to ask if that discrepancy between the calculated and observed wall temperatures would be explained on the basis of the assumed film factor; by the effect of the paper or something to do with the surface itself.

PROFESSOR KRATZ: Undoubtedly, that may be one thing. Our heat loss factors must be assumed, but they are stated under one condition only. In THE GUIDE they are estimated under conditions of a 40 deg mean temperature and a 15-mile wind. Variations in either one of those will affect the value of heat loss coefficient and thus affect the actual temperature as compared with the calculated temperature. They would also affect the calculated heat loss as compared with the actual heat loss. Undoubtedly variations in the film coefficient are the largest variations that enter into the problem.

MR. JOHNSON: I was thinking particularly of the film on the room side. Did you not make any attempt to evaluate that?

PROFESSOR KRATZ: No, I would be inclined to think that the larger variations would occur in the film on the outside, since the wind velocities are much more variable than the equivalent velocity produced by convection.

PRESIDENT FLEISHER: I would like to know whether the wind velocity was measured during the tests, and what the difference in effect would be on your insulated walls or your windows with window stripping and with different wind velocities. That might make a very great difference in your insulation.

Also, whether the sun effect had been taken into consideration, which would necessarily be less with your insulated walls than uninsulated, and whether with double windows you would lose the effect of radiant conditions outside which might affect your total over-all efficiency.

PROFESSOR KRATZ: Unquestionably, all those factors enter into this picture. We have data from our local bureau on the wind velocities and on the sun effects, but those are out in a field on the University farm about a mile and a half from the Residence. Any attempts that we have ever made to correlate our performance at the Research Residence against either wind velocities or sun effect have not been very successful. We did make some attempt in the uninsulated house, and did show that there was an effect produced by the wind and the sun, but we did not undertake to separate those effects in these tests at all.

F. E. GIESECKE: That is a very important one and needs additional and careful study. While it is called *film coefficient*, it includes the transfer of heat by radiation, conduction, and convection; there must be a considerable variation in this combined transfer of heat depending upon whether the coefficient is applied to floors, ceilings, high walls, or low walls. I believe the Committee on Research should plan a careful study of the *film coefficient* and its value for various surfaces and for various locations.

PROFESSOR KRATZ: Also, with regard to the *film coefficient*, the conception is based entirely on the assumption that the objects inside the room or outside, wherever they are, are at the same temperature as the air. Hence, if you have any surfaces that are either warmer or colder than the air temperature in the neighborhood of the surface, the *film coefficient* given in THE GUIDE does not apply at all, because you have changed the proportions of convection and radiation. Is not that true?

CECIL JONES: In regard to your wall temperature difference as compared with your room temperature difference, you might consider that the same as panel cooling and I would be interested in knowing the effect on the human being in the room when the wall was 15 F lower than the air temperature.

PROFESSOR KRATZ: We had no record of that. We had only a very small number of subjects, limited to visitors and the men on our staff, so we made no attempt to get a statistical study of comfort by keeping a record of the comfort of the different people that came in. I would say that the impression of the staff was that the house was more comfortable after insulation than before, and that was particularly true

in the third story rooms. The boys up there who were taking care of the readings and lived in the upstairs rooms reported a very great increase in the comfort, and it was not due to the increase in air temperature as much as getting rid of the cold surfaces around them.

H. H. DOUGLAS: We have conducted some tests in the south on insulation where it was possible to use electricity for heating. Due to the fact that the heaters were located directly in the rooms, this study originated in the form of a rate study. We took 32 scattered homes ranging from the small five-room home to the 30-room movie magnate's home. They were in operation from between four and eight years so that we eliminated the cold weather when the family might have been back East or the year when grandmother was here from the East, and consequently we felt that the results were quite accurate. We measured the power carefully in these homes, and they averaged 8 rooms to the home, and we obtained the information required by the rate department. One of our field men said, "I bought some of those insulated homes in my district and I am quite proud of them." He had sold the jobs so he suggested that we see if we could not deduct some conclusions from the amount of data assembled. Consequently, we went hastily through this list of 32 homes and found that exactly one-half of them were insulated homes and the remaining uninsulated. Therefore, we re-juggled this kilowatt hour input between the two different groups with the result that we found there was a difference in cost of operation of exactly 45 per cent, which confirms Professor Kratz's figures very nicely.

We also made a study with reference to the size of the homes in the different groups and found them so nearly alike in size that there was no need of any corrections.

In contrast to the Research Residence, however, our homes in Southern California are not built as a cube, a block; they are scattered all over half an acre, sometimes *U* shaped homes built with two or three or more different wings and angles, and consequently the outside exposure, which is the determining factor on heat losses is tremendously large in proportion to the volume as compared to the Research Residence. Another factor that is vastly different from Professor Kratz's study; in Southern California we think little of heating down there; in fact the early settlers from the East depended on Southern California sunshine, the weather, and the Chamber of Commerce to keep them warm. However, the Iowans and Minnesotans came out in increasing numbers and they found they needed heat and consequently there was an increased trend for better heating equipment.

The radiant effect of our California sunshine is a factor that we cannot evaluate in these studies. Another thing, and this is difficult to determine, is change due to heating conditions. We have a variation between maximum and minimum in Southern California quite frequently of 40 F during the day or night and, many times when the old Southern California sunshine comes out, we want to get rid of the heat as quickly as we got it, so that electricity lends and adapts itself to our climatic conditions. Therefore, we feel that the use of electricity for fuel, in view of the fact that we can measure the input so accurately, makes a study of this kind at least interesting, and we only wish that we had an opportunity to co-operate with the Research Laboratory on some of those installations that we have down there. It may not even be too late, because there are still some 4000 homes in Southern California that are heated with electricity.

PROFESSOR KRATZ: I might say that we realize that our results are not of laboratory accuracy. By separating out the different factors and treating each one separately we could obtain accuracy more nearly approximating laboratory accuracy. On the other hand, the thing that we were interested in was not so much laboratory accuracy regarding the various factors, as what is the net result when mother nature combines all of these factors with any kind of a whim she is inclined to take, and puts them in conjunction with a house which you already have, and in which you can use two different degrees of insulation and study under comparable conditions.

In other words, what is the integrated effect of all these factors in a particular case? That is the thing we were interested in, and I think that is about the only way that you can view these results. The answers must be viewed in the light of what we started out to do.

PRESIDENT FLEISHER: I again want to bring out the fact that the Society for years has been attempting to evaluate the exposure effect. Now we have been telling the contractor or the owner to guess at that particular factor and I think if the work that Professor Kratz has been doing is continued maybe we can finally establish a value for wind velocity on different exposures. It is certainly very essential.

S. H. DOWNS: Is it correct that the calculated heat losses for the house are higher than the experienced losses during the tests?

PROFESSOR KRATZ: The calculated heat losses at the design temperature difference were between 10 and 15 per cent higher than the actual, as determined from the fuel input and the losses in the flue gases.

MR. DOWNS: In Fig. 9 of the paper, the difference between the calculated and experienced losses is in the opposite direction. Is that correct?

PROFESSOR KRATZ: Yes, in that the actual temperature difference indicated that the actual wall was poorer than that assumed for the calculations.

MR. DOWNS: How do you account for the opposite directions of the differences in the two instances?

PROFESSOR KRATZ: There is no inconsistency represented because the temperature conditions apply only to the walls, while the heat losses from the house include losses due to infiltration, glass, etc. In the latter case it is apparent that the calculated losses from the walls alone were less than the actual. But the calculated losses from glass and infiltration were relatively much too great, and the net result was a total calculated heat loss from the house that was greater than the actual.

G. L. TUVE: In regard to the measurement of temperature in rooms in the Research Residence, I notice that the insulation seems to have little effect on the floor-to-ceiling temperature differential. I am a bit surprised at this. You said people would be equally comfortable, and I should like to know about the method of temperature measurement. Was the temperature measured at the middle of the room only, and with thermometers or with thermocouples? Did you use a set of measuring devices in each room? We are inclined to say the temperature of a room is 72 F, but if the thermometer is moved it registers 71, 73 or 74 F, because of the proximity of windows and cooled or heated surfaces and air currents. I know you took that into account, but this is a question on the general acceptance of room temperature measurements. We know the reading depends on where you put the thermometer or thermostat. Some people believe very much in taking room temperature on the basis of the thermostat location. I recently built a house and at first put the thermostat in the wrong place, and I know that it is a pretty tricky proposition. Should we not emphasize the importance of a low temperature gradient from inside wall to outside wall as well as studying the gradient from floor to ceiling? From your tests, it seems that the floor-to-ceiling gradient depends on the heating system only, and not on the wall construction.

PROFESSOR KRATZ: You opened up a rather broad subject, but to answer it as briefly as I can, we measure all of our temperatures in the Research Residence during the heating season by using carrying thermometers at different levels in the center of the room. This is the temperature on which we base these indoor-outdoor temperatures. We found from a study that we do get some variation between the temperature in the center and in the four corners of the room, but the variation is not over about 1½ deg. If you take readings at any one of those points you get the same characteristic variation up through the room as you do when you measure the temperatures up through the axis. Therefore, we have used the temperature taken up through the central axis by thermometers in order to obtain average values. If we want to know what is going on during a cycle we isolate one room and use thermo-

couples to observe the rapid temperature variations. All of our curves are based on readings of thermometers in the central axis of the room at the 60-in. level and on the average of 11 rooms so measured.

S. R. LEWIS: The architects are putting great pressure on us to do panel heating using warm air. Why not? It appears that much of the effect that has been observed in even the Research Residence has to do with panel heating after all. Increasing the efficiency of the insulation in the ceiling is desirable in order to use hot water coils imbedded in the plaster of the ceiling. We can do the same thing with warm air introduction near the ceiling, returning the air also near the ceiling and by this means a great deal of panel heating effect will be accomplished by radiation from the ceiling. I do not know how much. I am going to know more about that and I think that sort of thing should have additional study.

I remember Dr. Willard saying to me many years ago, "If you increase the efficiency of the insulation in the ceiling, thereby whatever is above that ceiling is going to be much colder, and the temperature difference between what is below the insulation and that which is above the insulation is going to increase." Thus while the heat transmitted through the insulation per degree temperature difference between the bottom and the top of the insulation is reduced, the increased number of degrees difference in temperature offsets this to some extent.

Dr. Giesecke and I have made some study of the drafts and down-flowing air currents along the windows in rooms that are thoroughly insulated and are heated with 130 F plaster in the ceiling. In my office which has the conventional radiator under the window, there is a very strong down current along the glass. That undoubtedly is caused by the air flow coming down from above to replace the up-draft above the radiator. We went on the same day to a room that was insulated, which had an enormous glass area, and which had no radiators, and no warm air supply register, but which was heated by a ceiling at about 120 deg. We were able to detect hardly any down draft along the glass. Grandmother could sit in front of the window and not get cold feet.

We may be approaching the remarkable comfort that may be evolved by using large heat transmitting areas at low temperatures by radiation from overhead.

MR. DOUGLAS: I wanted to bring up one point in connection with that fly-wheel effect or matter of storage involved in California and our South Coast country. The changes mentioned are so great between day and night that the fly-wheel effect becomes very apparent. We have one job in our San Bernardino commercial office where we installed a reverse cycle heating and cooling plant. This replaced an electric heating installation in service for 10 years. We know exactly the kilowatt hours used on this job and, consequently, since we installed the reverse cycle job, we have been able to compare the effect of reverse cycle against direct heat. Now the calculated heat loss on that building as far as cooling was concerned was 22 tons of refrigeration. We do not have a budget to put in 22 tons, so naturally as in the case of many utilities, we have shaved it down to 10 tons, leaving space for a 10-ton unit. That 10-ton unit carries the job perfectly satisfactorily. I am going to leave it up to Professor Kratz to discuss the fact that we have that tremendous variation from 55 F and 60 F to 100 F in the daytime and it is a concrete building with 1-in. walls and the storage effect, or fly-wheel effect on that building is tremendous.

About 10 years ago I had an opportunity to work in California Tech on a little research study on insulation. Various insulations were tested, among which we had redwood, one that can be obtained quite readily here on the Pacific Coast. Dr. Burke became so interested in that, that when he built his new home in Pasadena he used 8-in. studding and filled the walls with redwood sawdust. That made virtually an ice box out of the home and he tells me that his home is 10 or 15 deg cooler and easier to heat in the winter time. Just half of the fuel is used than before. He says all the old ladies in the neighborhood come to hold their quilting parties and they have no trouble entertaining their guests comfortably.

No. 1186

THE INTERACTION CONSTANT FOR MOIST AIR

By JOHN A. GOFF* AND A. C. BATES,** PHILADELPHIA, PA.

Preliminary report of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Towne Scientific School, University of Pennsylvania.

INTRODUCTION

IN Chapter 1 of THE GUIDE 1941 it was explained that an accurate and thermodynamically consistent formulation of the properties of moist air awaits the experimental measurement of a certain temperature function A_{aw}, called the interaction constant, expressing the effect of forces between air and water molecules. It is realized, of course, that there is no such thing as an air molecule, since air is itself a mixture of several gases; but the justification for regarding this mixture as a single constituent was also explained in THE GUIDE.†

The present paper is a preliminary report on the experimental investigation being conducted at the Towne Scientific School, University of Pennsylvania, under a cooperative agreement with the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS through its Research Technical Advisory Committee on Psychrometry.‡ The immediate objective of the investigation is the measurement of the interaction constant A_{aw} over the range of temperature of interest to air conditioning. To obtain satisfactory results by the method employed it was necessary to design a rather elaborate apparatus with which measurements could be made to an accuracy of at least 0.1 per cent. Satisfactory results have been obtained for only one temperature, namely 15 C, but it is felt that a preliminary report outlining the theory of the experiment, describing the apparatus, and discussing present results will be of interest.

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** Assistant Professor of Mechanical Engineering, Towne Scientific School, University of Pennsylvania.

† In this and subsequent references to THE GUIDE, Chapter 1 of THE HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1941 is intended.

‡ *Committee Personnel:* F. R. Bichowsky, *Chairman;* D. B. Brooks, W. H. Carrier, H. C. Dickinson, A. W. Gauger, John A. Goff, William Goodman, A. M. Greene, Jr., L. P. Harrison, F. G. Keyes, A. P. Kratz, D. M. Little, Axel Marin, D. W. Nelson, W. M. Sawdon and F. O. Urban.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, San Francisco, Calif., June, 1941.

THEORY

The method employed may be described briefly as the *saturation isotherm* method. A steady stream of dry, carbon dioxide-free air is passed through a saturator at high pressure P_1, then through a drying train in which m_1 pounds of water is removed, then through a second saturator at low pressure P_2 and finally through a second drying train in which m_2 pounds of water is removed. Both saturators are immersed in the same thermostat whose temperature T is closely regulated and accurately measured. Assuming (1) complete saturation in the saturators, (2) complete drying in the drying trains, (3) negligible leakage, (4) correctness of theory, and (5) accuracy of all measurements, the five data P_1, P_2, m_1, m_2, T suffice for the determination of the interaction constant A_{aw} without requiring to weigh the quantity of dry air passed through the apparatus.

In order to provide a check on the above assumptions, it was arranged to weigh the dry air. Preliminary saturators in a separate thermostat were provided so that saturation in the main thermostat could be approached either by condensation or by evaporation. Suitable reducing valves were provided to permit changing the pressures P_1 and P_2 in order to make sure that the results are independent of pressure and dependent only on temperature as required by theory. Orifices of different sizes were provided for the discharge line in order to be able to vary the rate of flow through the apparatus.

The theory of the experiment is built upon the prediction of statistical mechanics expressed in Equation (9) of THE GUIDE, namely,

$$Pv = RT - [A_{aa}x^2 + 2A_{aw}x(1-x) + A_{ww}(1-x)^2]P \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

P = observed pressure
v = specific volume (volume per mol)
A_{aa} = second virial coefficient for dry air, a known temperature function
A_{ww} = second virial coefficient for water vapor, a known temperature function
A_{aw} = the interaction constant, the temperature function to be measured
x = mol fraction of dry air in the mixture

Applying a well-known identical relation of thermodynamics to (1) and observing a further prediction of statistical mechanics in adjusting the constant of integration, the following expressions for the chemical potentials are easily derived:

$$\mu_a = RT \log_e (xP) + [(A_{aa} - 2A_{aw} + A_{ww})(1-x)^2 - A_{aa}]P + g_a(T) \quad . \quad (2a)$$

$$\mu_w = RT \log_e [(1-x)P] + [(A_{aa} - 2A_{wa} + A_{ww})x^2 - A_{ww}]P + g_w(T) \quad . \quad (2b)$$

where $g_a(T)$ and $g_w(T)$ are temperature functions which will not need to be evaluated explicitly.

As explained in THE GUIDE (p. 18), the conditions for saturation are equality, between the two coexisting phases, of (a) pressure (b) temperature and (c) each component chemical potential. Therefore, in order to set up the equation for the saturation isotherms of moist air, suitable expressions for the chemical potentials of dry air and water in the liquid phase must first be written down.

Now the saturated liquid phase will be almost pure water, but containing a very small concentration of dissolved air. This dissolved air will not have the

same composition as that of dry air, but will show considerable oxygen enrichment. However, this effect is entirely negligible and it is legitimate to regard the liquid phase as an *ideal solution* of dry air in water. Furthermore, in the range of pressures investigated the liquid phase may also be regarded as incompressible; whence the appropriate expressions for the chemical potentials are [1]

$$\mu'_a = RT \log_e x' + \overline{V}'_a P + g'_a(T) \quad (3a)$$

$$\mu'_w = RT \log_e (1 - x') + v'_w P + g'_w(T) \quad (3b)$$

where

x' = mol fraction of dissolved air
$\overline{V}'_a$ = partial molal volume of dissolved air
v'_w = specific volume of pure liquid water

and $g'_a(T)$, $g'_w(T)$ are temperature functions which will not need to be evaluated explicitly. The prime refers to the liquid phase.

Equating (2b) and (3b) as one condition for saturation, and then noting that x and x' must vanish simultaneously when the pressure P is reduced to the saturation pressure p_s of pure water for the given temperature, the result is

$$RT \log_e \left[\frac{(1-x)P}{p_s}\right] = RT \log_e (1 - x') + (v'_w + A_{ww})(P - p_s) - [(A_{aa} - 2A_{aw} + A_{ww})x^2] P \quad (4a)$$

Next, equating (2a) and (3a), as a second condition for saturation,

$$RT \log_e \left[\frac{x'}{xP}\right] = -\overline{V}'_a P + [(A_{aa} - 2A_{aw} + A_{ww})(1-x)^2 - A_{aa}] P + g_a(T) - g'_a(T) \quad (4b)$$

The mol fraction x' is to be eliminated between these two equations in order to obtain the relation between mol fraction x and pressure P for a given temperature.

Fortunately, the required elimination can readily be effected by the use of approximations which can be validated later. In the first place, the term $RT \log_e (1 - x')$ may be replaced by $-RT\,x'$ since x' is so very small; and, this being so, the actual dependence of the righthand member of (4b) on pressure and mol fraction x may be ignored and this equation reduced to the form of Henry's Law for dilute solutions, namely,

$$\frac{x'}{xP} = k(T) \quad (5)$$

Finally, it may be anticipated that the righthand member of (4a) will be small enough to justify the use of the approximation, $1 - x = p_s/P$, for the purpose of evaluating (5) and the rightmost term of (4a). The final result after dividing through by RT, is

$$\log_e \left[\frac{(1-x)P}{p_s}\right] = A\left(\frac{P}{p_s} - 1\right) - B(1-\lambda)\left[\frac{P}{p_s} - 2 + \frac{p_s}{P}\right] \quad (6)$$

where

$$A = -kp_s + \frac{v'_w p_s}{RT} + \frac{A_{ww} p_s}{RT} \quad (7a)$$

[1] Numbers refer to references listed in Bibliography.

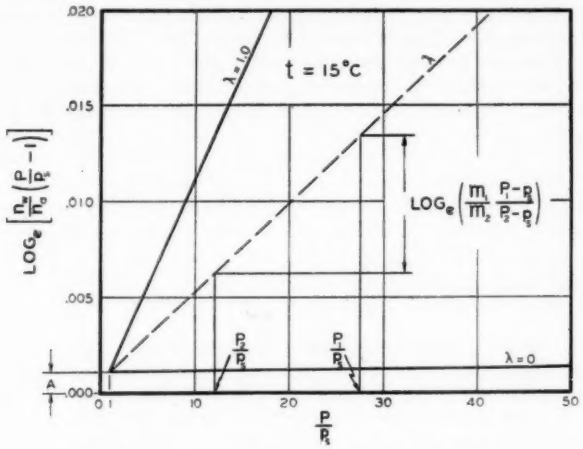

FIG. 1. GRAPHICAL DETERMINATION OF *Lambda*

and

$$B = \frac{A_{aa}p_s}{RT} + \frac{A_{ww}p_s}{RT} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7b)$$

are dimensionless temperature functions which can be calculated from data available in the literature; and where

$$\lambda = \frac{2A_{aw}}{A_{aa} + A_{ww}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

is introduced as a dimensionless interaction constant which, according to theory, should depend on temperature only. In what follows, attention will be directed to the determination of λ rather than of A_{aw} itself.

It will be noticed that, if λ were known for a given temperature, the mol fraction x could be calculated from (6) for various values of pressure P. From the values thus obtained, values of mol *ratio*, $n_w/n_a = (1-x)/x$, could be calculated and multiplied by $(P/p_s - 1)$. In this manner a family of curves of which three are shown in Fig. 1 could be determined for any given temperature. Suppose, then, that the *same* weight of dry air n_a (mols) were saturated first at pressure P_1 and then at lower pressure P_2 but the *same* temperature. The difference between values of the ordinate in Fig. 1 between the two pressures would be

$$\log_e \left[\frac{m_1(P_1 - p_s)}{m_2(P_2 - p_s)}\right],$$

where the weight of air cancels out and m_1 and m_2 are the weights of water (in pounds or grams) removed after each saturator. But this difference in ordinate can be measured directly: hence the determination of λ for the given temperature reduces to picking that curve of Fig. 1 showing the measured

difference in ordinate between the measured pressures. Actually, this procedure was formulated mathematically so that values of λ could be calculated directly without resort to graphical representation. This was an important step in expediting routine calculations, but need not be given here.

It is desired to emphasize that the above method of determining λ does not require to know the weight of dry air passing through the apparatus. However, the weight of dry air was actually measured so that measured values of mol fraction x were available for substitution in (6) as a check. At first this check failed by an amount which could be attributed to an error of about 0.08 C in the measurement of temperature. The thermometer used had been calibrated by the Bureau of Standards in 1924; but, since corrections were given at 20-degree intervals only, it was necessary to interpolate for a correction at 15 C. While the thermometer was being recalibrated at the Bureau of Standards, the apparatus was carefully examined for leaks only to find them entirely negligible; also minor changes were made to improve pressure regulation. When the thermometer was returned after recalibration, the actual correction at 15 C was found to be almost exactly what was required to provide a check with the measured weight of dry air through (6). This was most gratifying.

EXPERIMENTAL METHOD AND APPARATUS

The choice of experimental method, for which the theory of the preceding section was developed, was based on a study of the difficulties likely to be encountered in alternative methods—such as measuring simultaneous values of pressure, specific volume, temperature and composition and calculating A_{aw} directly from (1)—and the consideration that a direct determination of the weight of water per pound of dry air *at saturation* would be of special interest to the air conditioning engineer. Before attempting to design the apparatus, however, it was necessary to decide what range of temperature was to be covered, at what pressures it was feasible to operate, what precision in each measured quantity would be required in order to obtain significant values of λ.

The lower limit of temperature was easy to decide upon as 0 C, since below this the condensed phase would be ice which would dictate a radically different saturator design. The upper limit was taken to be 30 C. Preliminary consideration of a curvature of the saturation isotherms at low pressures according to (6) suggested $P_2 = 10p_s$ as lower limit for pressure. The upper limit was taken to be $P_1 = 200\ p_s$. Keyes [2] had estimated that at atmospheric pressure saturated air at 30 C and at 40 C would contain about 1 per cent more water vapor than the ordinary application of Dalton's Law would predict. This made it appear necessary not to allow any measured quantity to be uncertain by more than 0.1 per cent. The severity of this precision requirement will be the better appreciated when it is realized that at 0 C, 10 p_s is only 1.85 in Hg and 0.1 per cent of this is only 0.002 in Hg. In order to attain accuracy of this sort, automatic control and regulation of temperature and pressure over comparatively long test periods and painstaking attention to all details were absolutely necessary. The design calculations, the problems of matching and compromising equipment characteristics, and preliminary tests on component parts of the apparatus cannot be discussed in this report for lack of space. Suffice it to say that the demands for inventiveness, physical intuition.

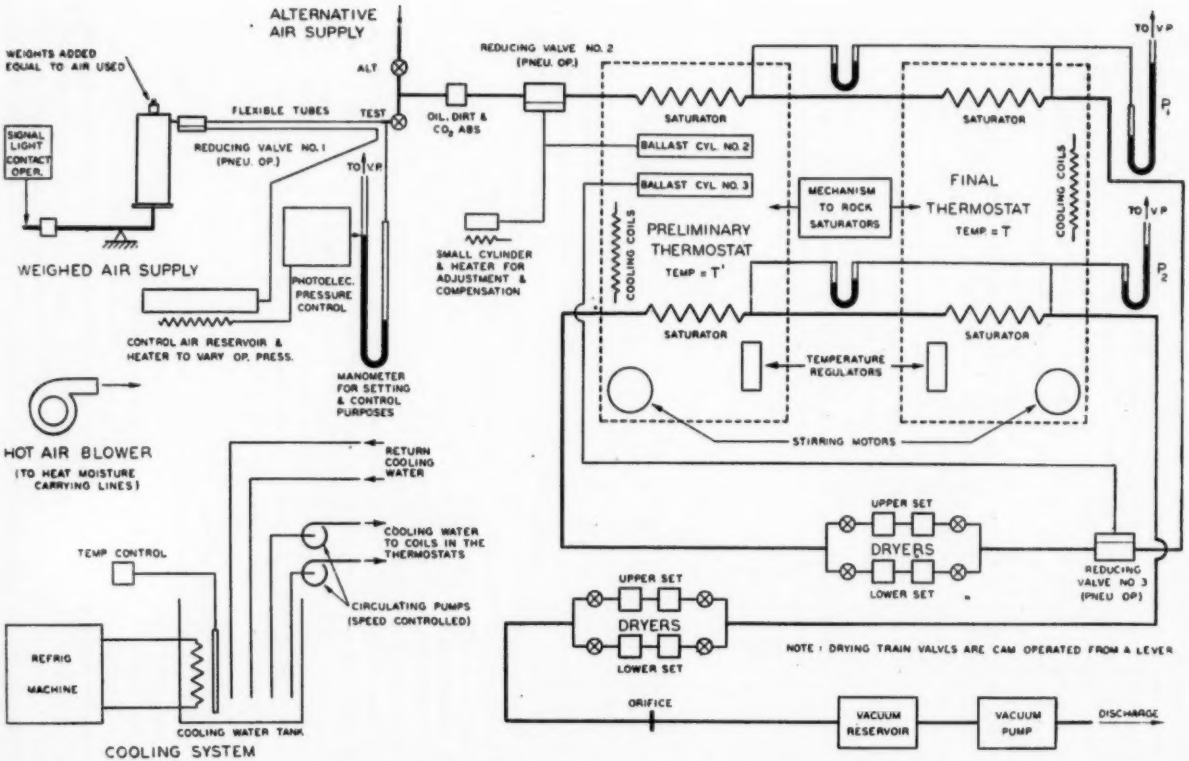

FIG. 2. SCHEMATIC DIAGRAM OF THE TEST EQUIPMENT

and mechanical skill were considerable and explain the apparent slowness in obtaining even preliminary results.

Fig. 2 is a flow diagram of the apparatus in its present form and represents an attempt at completeness without loss of simplicity. The photographs of Figs. 3 to 6 give an idea of appearance, arrangement, proportions and some details.

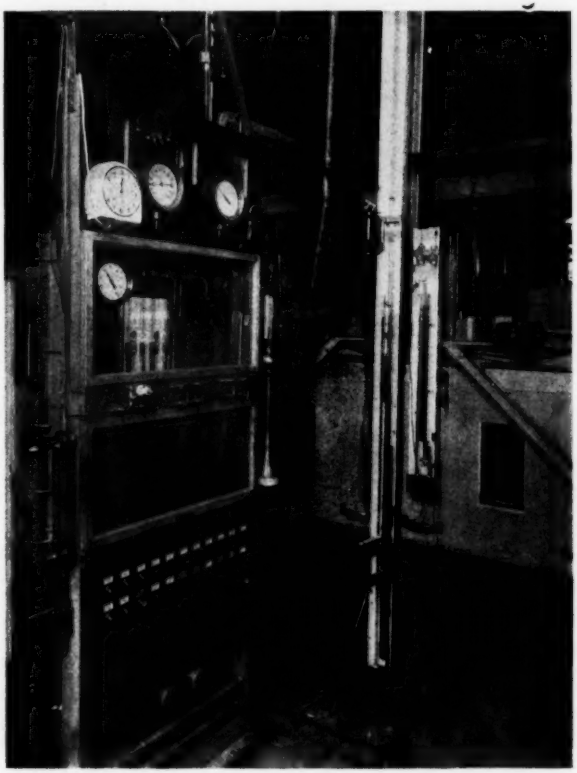

FIG. 3. GENERAL VIEW

During actual test periods the supply of air comes from a small cylinder initially containing from 150 to 220 grams of air depending on how many times it has been filled from a standard 200 cubic foot cylinder. When it is no longer possible to fill the small cylinder to a pressure of 1500 lb per square inch, a fresh large cylinder is used, the air remaining in the old one being used for pre-test and in-between runs, for charging the pneumatic controls, and for general utility purposes.

FIG. 2. SCHEMATIC DIAGRAM OF THE TEST EQUIPMENT

The small test cylinder depends from one arm of a sensitive balance. In order to limit the pressure in the take-off tube and keep it reasonably constant, a small reducing valve was mounted directly on the cylinder. This valve was modified for pneumatic control from the main control panel, which obviated the necessity of disturbing the balance except to add weights, but required an

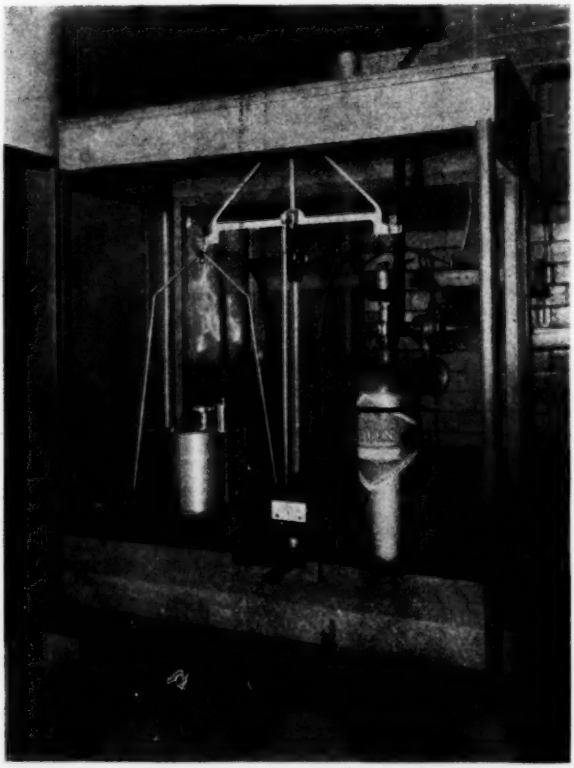

FIG. 4. WEIGHED AIR SUPPLY

additional tube paralleling the take-off tube. Weights were added directly over the center of gravity of the cylinder so as not to alter the position of the cylinder relative to the tubes. In order to eliminate correction for buoyancy, a dummy weight made of aluminum and having about the same density as that of the cylinder and its contents was placed on the other scale pan. Although some minor improvement can yet be made in the air weighing apparatus, nevertheless, in its present form it has been proven not to introduce an error of as much as 0.1 per cent of the total weight of air withdrawn during

the test period. As suggested above, weighing was done by the method of substitution, balance being indicated by electrical contact.

Owing to the fact that the pressure in the test cylinder drops continually as air is withdrawn during test, the first reducing valve requires some compensation. To provide this a mercury manometer was connected to the delivery

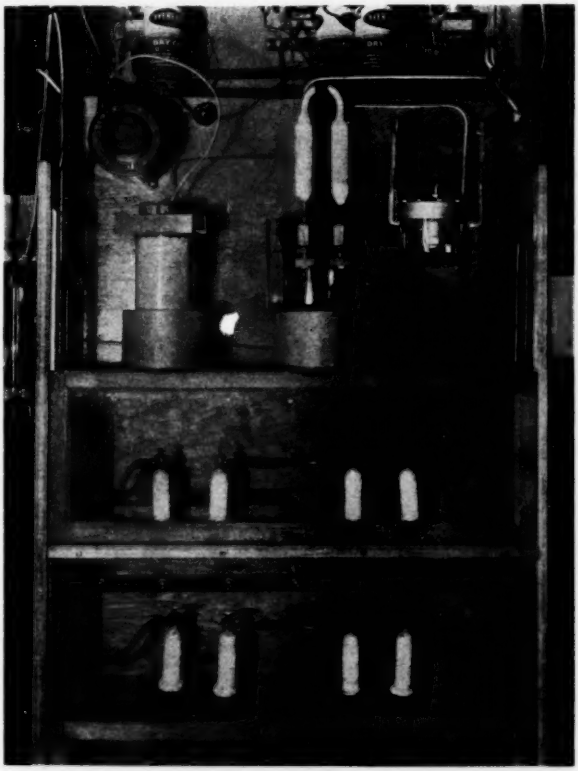

FIG. 5. DRYERS, BACK OF PANEL

line. On the manometer was mounted a photoelectric cell which operated through a suitable relay to heat the air on the control side of the valve thus raising the control pressure and compensating for the drop in supply pressure.

The air delivered by the first reducing valve passed through a CO_2 remover, then through a second reducing valve, also pneumatically controlled, then through the high pressure saturator in the preliminary thermostat to the high pressure saturator in the main thermostat. The second reducing valve was for the purpose of closer pressure regulation.

The saturators were patterned after those used by Gordon [3] and Washburn [4] but were constructed from ½ in. O.D., 0.035 in. wall tubing with solder type fittings instead of glass, and modified by the addition of a filling manifold and gauge glasses. The use of copper was dictated by pressure and heat transfer requirements. After Test No. 3 the rocking speed of the saturators was re-

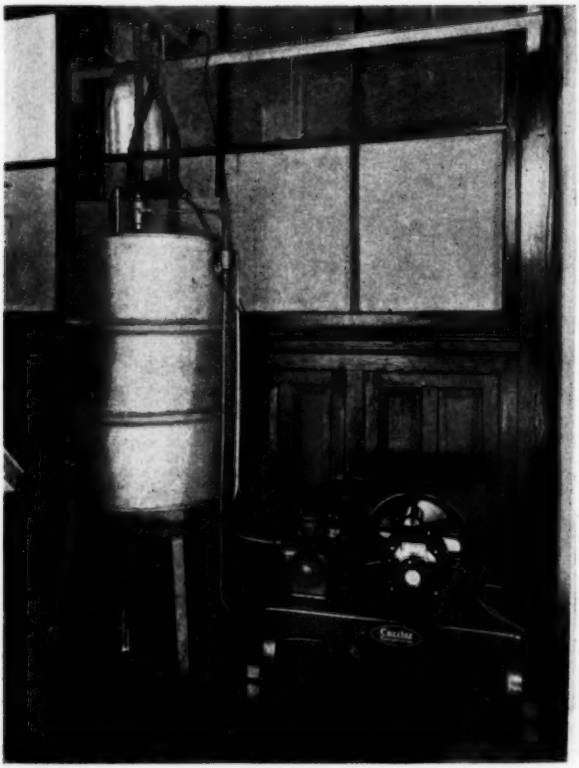

FIG. 6. COOLING SYSTEM

duced to about once in forty seconds it having been noted that, at the higher speed of once in fifteen seconds used in previous tests, slight pressure pulsations occurred due to the formation of a film causing momentary interruption of the flow. Connections to the saturators were made of copper tubing designed to be leak proof and self draining, with adequate fatigue life. All lines to and from the saturator were heated by air from an ordinary hair dryer to prevent condensation. This warm air was carried over to the glass-enclosed control panel where various valves carrying moist air were located.

In all tests after Test No. 3, anhydrone was used as drying agent. The usual procedure of providing alternative drying sets of approximately matched flow resistances, switching from one set to the other at start and finish of each test, was followed. The switch-over valves were cam operated from a single cam shaft, care being taken not to turn the shaft too quickly for fear of carrying some of the finely divided anhydrone out of the drying towers. Other precautionary measures were taken in the handling of the dryers which need not be described in detail here.

TABLE 1—TEST DATA

1	Test Number	2	3	4	5	6	7	9
2	Temperature, main thermostat (deg C)..	14.91	14.91	14.91	14.91	14.91	14.91	14.91
3	Temperature, preliminary thermostat (deg C).....	14.9	20.	14.9	14.9	14.9	14.9	14.9
4	High Pressure (in. Hg).	50.522	50.573	50.630	50.413	50.490	25.395	25.272
5	Low Pressure (in. Hg).	25.266	25.272	25.265	25.261	25.356	12.560	12.594
6	Weights added to scales (grams).....	150	160	115	170	170	85	170
7	Duration of Test (hours)	3.88	4.11	2.97	4.39	4.37	4.51	9.33
8	Nominal Rate of Flow (grams per hr).....	40	40	40	40	40	20	20
9	Weight water, first dryer (grams).....	0.9542	1.0099	0.7247	1.0744	1.0732	1.0733	2.1581
10	Weight water, second dryer (grams).....	1.9001	2.0254	1.4581	2.1567	2.1479	2.2115	4.4101
11	Corrections:							
	(a) Change in control press.....	+ 0.002	+ 0.005	+ 0.004	+ 0.006	+ 0.004	+ 0.001	+ 0.004
	(b) Change in delivery press.....	+ 0.017	+ 0.006	0.000	− 0.006	0.000	+ 0.026	0.000
	(c) Dirt, H_2O, CO_2..	− 0.090	− 0.096	− 0.069	− 0.102	− 0.102	− 0.051	− 0.102
	(d) Leakage.....	...	...	− 0.018	− 0.026	− 0.026	− 0.027	− 0.056
	(e) Net.....	− 0.071	− 0.085	− 0.083	− 0.128	− 0.124	− 0.051	− 0.154
12	Corrected weight of dry air.....	149.93	159.91	114.92	169.87	169.88	84.95	169.86

Returning to the flow diagram, the air stream from the high pressure saturator was sent through a third reducing valve before entering the first drying train in order to obviate the necessity of designing the towers for the high pressure. It then passed through the low-pressure saturator in the preliminary thermostat, then through the low-pressure saturator in the main thermostat, then through the second drying train, and finally discharged into an evacuated carboy through a suitable orifice to regulate the rate of flow.

For measuring the pressures at exit from the two main saturators, mercury manometers with one leg connected to a high-vacuum pump were used, the absolute pressure being measured by a McLeod gauge. This arrangement made the readings of the manometers independent of changes in barometric pressure. Excessive trouble was experienced in keeping the mercury menisci

clean until one-piece glass construction was adopted. It is felt that ultimate precision in the measurement of pressure has not yet been reached and some changes will be made before additional tests are started.

The temperature regulators were constructed along conventional lines. They operated through thyratron tubes which start and stop suitable pumps supplying refrigerated water (with a suitable antifreeze added to prevent freezing and improve heat transfer) to the cooling coils in the thermostats. The evaporation coil of the small refrigeration machine was placed in a separate insulated tank partially filled with the cooling medium. This arrangement not only made for good control, but allowed for the possibility of inserting an electric heater in experiments at higher temperatures. The temperature regulation at 15 C was about $\pm$ 0.001 C as determined by a Beckmann thermometer. Closer regulation, though obtainable, was not considered necessary.

The pneumatic operation of all reducing valves makes them insensitive to changes in barometric pressure, permits their use at pressures above or below atmospheric, and makes delicate adjustment and automatic control possible. The extent of the electrical auxiliaries may be indicated by mentioning that the apparatus includes twenty-two switches, two thyratron tubes, and one photoelectric cell plus a relay tube.

TEST PROCEDURE

A detailed explanation of the test procedure will be omitted from this report but will be found in the thesis § submitted by one of the authors to the Faculty of the Graduate School of Arts and Sciences, University of Pennsylvania, in partial fulfillment of the requirements for the doctor's degree in engineering. A copy is on file in the Towne Scientific School Library.

TEST DATA

Test No. 1 is to be rejected because of incomplete drying. Test No. 8 is also rejected because of unfortunate failure to record the initial weight of one of the drying towers. Table 1 lists the data obtained in all other tests so far performed. The temperature was the same in all tests, namely 14.91 C.

Changes in control pressure and delivery pressure alter slightly the quantity of air stored in the first reducing valve so that the loss of weight of the test cylinder assembly is not exactly the weight of air reaching the main apparatus. Special preliminary tests were performed to determine the necessary corrections (items 11a and 11b). The correction for H_2O in the supply air (item 11c) was computed by assuming that the air in the large supply cylinder is continuously saturated during the process of filling the small test cylinder. The correction for dirt and CO_2 was determined by passing a weighed sample of air through the CO_2 remover and noting the gain in weight (item 11c). Leakage corrections were inferred from pressure readings over extended periods under shut-off conditions (item 11d). For Tests 2 and 3, no leakage corrections were determined.

§ Doctoral Dissertation, University of Pennsylvania, 1941. "Measurement of the Interaction Constant for Moist Air by the Saturation Isotherm Method," by Arthur Crocker Bates.

REDUCTION OF TEST DATA

In reducing the data of Table 1, the thermal data given in Table 2 as obtained from the literature were used.

Values of the saturation pressure p_s are Bureau of Standards [5] values converted to inches of mercury at 68 F, $g = 980.196$ (Philadelphia). Values of kp_s were computed from solubility data in International Critical Tables. Values of $v'_w p_s/RT$ were computed from steam table data.[5] Values of $A_{ww}p_s/RT$ were calculated from the formula of Keyes.[6] Values of $A_{aa}p_s/RT$ were calculated from the data of Holborn and Otto [7] and from the formula of Beattie and Bridgeman,[8] as modified by Keyes.[2] The quantities A and B were defined in terms of the preceding quantities by Equation (7).

Table 3 shows the final reduction of the data of Table 2. Values of the interaction constant λ (item 6) are computed without using the measured

TABLE 2—THERMAL DATA FROM THE LITERATURE

t C	p_s (IN. HG)	kp_s x10^5	$v'_w p_s/RT$ x10^5	$A_{ww}p_s/RT$ x10^5	$A_{aa}p_s/RT$ x10^5	A x10^5	B x10^5
0	0.18110	1.42	0.485	0.5609	3.598	0.5656	0.5645
5	0.25853	1.83	0.676	0.7081	4.611	0.7147	0.7127
10	0.36385	2.25	0.94	0.8850	5.669	0.8942	0.8907
15	0.50529	2.83	1.28	1.0956	7.161	1.1081	1.1028
20	0.69290	3.55	1.72	1.3449	8.676	1.3618	1.3526
25	0.93884	4.53	2.29	1.6370	10.301	1.6594	1.6473
30	1.25774	5.51	3.06	1.9780	11.978	2.0080	1.9900
14.91	0.50236	2.82	1.28	1.0915	7.136	1.1042	1.0986

weight of dry air in the manner previously explained. With λ determined in this manner the mol fraction of water vapor $1 - x$) is computed from (6) for either high or low pressure (item 2 or item 3); then using the corresponding measured weight of water (item 4 or item 5), the weight of dry air is calculated (item 7). Item 9 is the difference between item 8 and item 7 expressed in per cent.

Little significance is assigned Tests 2 and 3 except as they indicate that saturation was being closely approached by showing that whether the preliminary thermostat was set for 15 C or 20 C had little effect. The reasons for discounting these tests are certain irregularities in the performance of the apparatus and unsatisfactory drying. In Test 4, it was forgotten to start the rocking motor at the beginning of the test; therefore, it was left off during the entire test in order to see the effect upon the results.

Tests 5, 6, 7, 9 are considered the only ones that should be used in computing the most probable value of λ. Item 10 shows that these four tests are remarkably consistent showing an average failure to cross-check the weight of dry air of only—0.08 per cent. This might be due to a residual temperature error of 0.012 C which could be merely an interpolation error since the thermometer was graduated only to tenths. On the other hand, there might

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TABLE 3—REDUCED TEST DATA

1	TEST NUMBER	2	3	4	5	6	7	9
2	High Pressure Ratio (P_1/p_a)	100.569	100.671	100.784	100.352	100.506	50.551	50.307
3	Low Pressure Ratio (P_2/p_a)	50.295	50.307	50.293	50.285	50.474	25.002	25.070
4	Weight water, first dryer (grams)	0.9542	1.0099	0.7247	1.0744	1.0732	1.0733	2.1581
5	Weight water, second dryer (grams)	1.9001	2.0254	1.4581	2.1567	2.1479	2.2115	4.4101
6	Interaction Constant	0.253	0.138	0.105	0.072	0.085	0.064	0.082
7	Calculated Weight Dry Air (grams)	148.33	159.14	114.74	169.99	169.82	85.09	170.08
8	Corrected Weight Dry Air (grams)	149.93	159.91	114.92	169.87	169.88	84.95	169.86
9	Per cent Deviation	+1.07	+0.48	−0.16	−0.07	+0.04	−0.16	−0.13
10	Average Deviation				−0.08%			
11	Drying Agent	Activated Alumina			Anhydrone			

be a consistent error in the air weighing; and in future tests some improvements calculated to reduce this uncertainty will be made. Also some indicated improvements in the measurement of pressure will be made since the present failure to check might be explained by errors in pressure of a few thousandths of an inch of mercury.

The variation in the values of λ for the last four tests as given in Table 3 might at first blush appear disappointing. However, it must be remembered that extreme precision would be required to reduce this variation below about 5 per cent. After careful consideration of all recognizable sources of error in the individual tests and weighting the results accordingly, it appears safe to say that

$$\text{at } 14.91 \text{ C}, \lambda = 0.075 \pm 0.005$$

REDUCTION OF POLLITZER AND STREBEL DATA

So far as the authors are aware, the only published data with which to compare the results of this investigation are those of Pollitzer and Strebel,[9] at 50 C and 70 C. These measurements are by no means of comparable precision with those presented here; but they yield fairly good values of λ because, at the much higher pressures employed, the precision requirements are much less severe. Thus the average departure from Dalton's Law for the range of pressure employed by Pollitzer and Strebel is about 30 per cent instead of 0.6 per cent for the range covered in the present experiments. Offsetting this advantage to an unknown extent, however, is the fact that the theory available at the present time is limited to the range of pressures in which Equation (1) is applicable.

Two methods were used in reducing the data of Pollitzer and Strebel. In the first, a value of λ was computed for each test using Equation (6). Assuming no systematic error, an attempt was made to select intervals of λ which

would give a normal frequency distribution. This can be done for the 50 C tests by disregarding low pressure points which are obviously erratic; but no choice of intervals could be found to make a normal distribution of errors for the 70 C tests. The most probable values of λ determined in this way are judged to be: at 50 C, $\lambda = 0.057 \pm 6$ per cent; at 70 C, $\lambda = 0.087 \pm 16.5$ per cent.

The second method is to apply least squares to determine λ and a factor ϵ by which the nominal saturation pressure p_s might be multiplied to account for a small error in measurement of temperature and/or weight of air. Again omitting low pressure points, the results are:

at 50 C, $\lambda = 0.052$ and $\epsilon = 1.024$; at 70 C, $\lambda = 0.065$ and $\epsilon = 1.033$.

After careful consideration the following results were selected as representing the best reduction of the Pollitzer and Strebel data possible without employing methods involving computational labor entirely unjustified by any use to be made of the results

50 C: $\lambda = 0.056$
70 C: $\lambda = 0.07$

Fig. 7 shows the desired comparison with present results. No curve is drawn, but there is some indication that λ does not vary markedly with temperature and may be assumed to have a constant value of about 0.075 pending further experimentation.

DISCUSSION OF RESULTS

If the object of the present investigation were merely to adjust λ so as to make Equation (6) fit the experimental data, very little discussion beyond that already given would be required. However, it is proposed to use the values of λ inferred from these experiments, and from those at different temperatures to follow, for the purpose of computing specific volume by means of Equation (1) and specific enthalpy from the following equation obtained by applying a well-known identical relation of thermodynamics to Equation (2):

$$h = xh^\circ_a + (1 - x)h^\circ_w + [B_{aa}x^2 + 2B_{aw}x(1 - x) + B_{ww}(1 - x)^2]P \quad . \quad . \quad . \quad . \quad (9)$$

where

h°_a = zero-pressure specific enthalpy of dry air obtained from spectroscopic data (Table 3, THE GUIDE)

h°_w = zero-pressure specific enthalpy of water vapor obtained from spectroscopic data (Table 5, THE GUIDE)

and where, for example,

$$B_{aw} = -T^2 \frac{d(A_{aw}/T)}{dT} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

Therefore it is important to adduce as much evidence as possible for believing that the value of λ determined from (6) is indeed the interaction constant to be used in (1) and (9).

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TABLE 4—THERMODYNAMIC PROPERTIES OF MOIST AIR, 29.921 IN. HG

TEMP DEG F	SATURATION HUMIDITY RATIO W_s WEIGHT OF WATER PER LB OF DRY AIR		VOLUME CU FT PER LB DRY AIR			TEMP DEG F
	Pounds	Grains	Dry Air v_a	v_{as} $(v_s - v_a)$	Saturated Mixture v_s	
68	0.01468	102.8	13.29	0.31	13.60	68
68	*0.01477*	*103.40*	*13.299*	*0.314*	*13.613*	*68*

TEMP DEG F	ENTHALPY BTU PER LB DRY AIR			SPECIFIC ENTHALPY OF LIQUID WATER BTU PER LB h'_w	SATURATION PRESSURE p_s		TEMP DEG F
	Dry Air h_a	h_{as} $(h_s - h_a)$	SATURATED MIXTURE h_s		In. of Hg	Lb per Sq In.	
68	16.31	16.00	32.31	36.0	0.68980	0.3388	68
68	*16.324*	*16.089*	*32.413*	*36.04*	*0.69413*	*0.3409*	*68*

In the first place, it is important to adduce evidence of the complete saturation assumed in (6) or, alternatively, evidence of independence of the final results on whether complete saturation is attained or not. For example, theory [10] indicates that, at constant temperature, the degree of saturation depends only on the weight of air flowing through the saturators per unit time. The weight flowing through each pressure stage being the same in any one test, the same degree of saturation should be reached in both saturators and the ratio m_1/m_2, on which the determination of λ depends, should be correct even if complete saturation were not attained. Of course, incomplete saturation would cause failure to cross-check with the measured weight of air through (6); however, the average deviation of -0.08 per cent is of the wrong sign to be explained by incomplete saturation, though this fact in itself does not completely exclude the possibility. The higher rate of flow in Tests 5 and 6 would be expected to result in lower values of $(1 - x)P/p_s$ for these tests than for Tests 7 and 9. To assume this effect just sufficient to bring the points for the former into agreement with those for the latter at $P/p_s = 50$ would amount to assuming that doubling the rate of flow lowered the degree of saturation by 0.1 per cent. Making this assumption would reduce somewhat the probable error in λ, but at the same time would increase numerically the average deviation between measured and calculated weight of air. Additional tests with the preliminary thermostat at higher temperature, plus tests at a third rate of flow, might explain present discrepancies. Therefore the limits of uncertainty in the value of λ have been left as determined without indulging in assumptions.

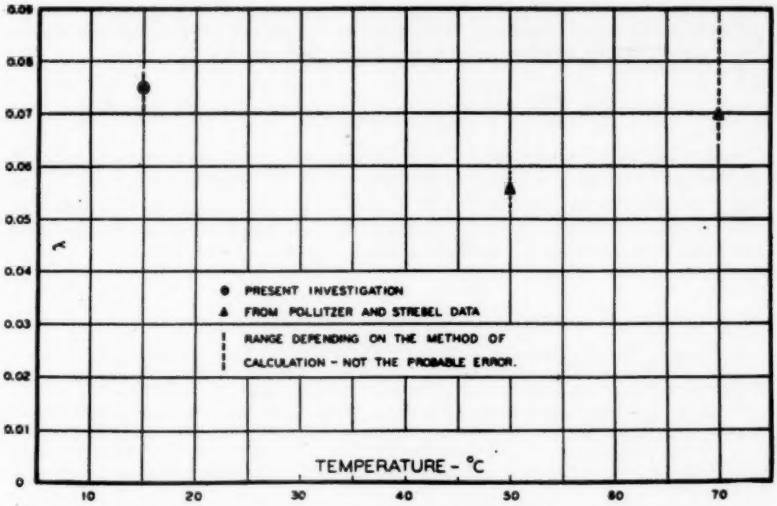

FIG. 7. AVAILABLE VALUES OF *Lambda*

CONCLUSIONS

The following conclusions are considered reasonable: (1) the theory is adequate, at least in the low-pressure range, since all deviations can be ascribed to residual experimental error; (2) the two-stage saturation isotherm method gives exceedingly consistent results; (3) weighing the air is practically indispensable as a cross-check until satisfactory operation of the apparatus is established; (4) the possibility of measuring λ by the saturation isotherm method is established; (5) aside from the precision of all direct measurements, the two most important experimental items are elimination of leakage and extremely close pressure regulation; (6) now that the theory has been checked in the low pressure range, future tests may be made at higher pressures where it is easier to determine λ accurately even though it may then be impossible to weigh the air on account of the limited capacity of the present weighing apparatus.

SUMMARY

Preliminary measurements of the interaction constant for moist air by the saturation isotherm method are reported. The value $\lambda = 0.075 \pm 0.005$ for 15 C is regarded as the best reduction of the data obtained. Comparison with certain data in the literature indicates that λ will be found not to vary markedly with temperature. This would correspond to a Dalton factor, at standard atmospheric pressure and 20 C, of 1.0052 which is much smaller than the value guessed in THE GUIDE 1941 but still much larger than the Poynting

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factor and very much larger than the Raoult factor, two corrections usually applied to the vapor pressure. The indicated revisions of the data are shown in Table 4, (Table 6, Chapter 1, THE GUIDE 1941, at 20 C).

ACKNOWLEDGMENT

Acknowledgment is made to F. R. Bichowsky, Chairman of the ASHVE Research Technical Advisory Committee on Psychrometry, whose advice on experimental matters has contributed greatly to the success of the investigation; also to W. H. Carrier for his cooperation in lending a small refrigeration machine used in connection with this investigation.

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No. 1187

ECONOMICAL AIR VELOCITIES FOR MECHANICAL AIR FILTRATION

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MINNEAPOLIS, MINN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Engineering Experiment Station, University of Minnesota.

IN the removal of impurities from the air by mechanical filters of the non-automatic cleaning type, the question often arises as to what filtration air velocity should be adopted for the most economical operation. If a comparatively high air velocity is chosen, the initial cost of the installation is reduced, as fewer filters are required to handle the necessary volume of air. On the other hand, as the resistance of the air filters to air flow varies as some power (greater than one) of the air velocity, the useful life and dust holding capacity of the filtration system between recharging or replacement of the filters may be greatly reduced. Other complicating factors involved in the determination of the most economical velocity for air filtration, in addition to the cost of replacement or recharging of the filters and the yearly fixed charges or depreciation on the initial installation, are filtration power costs and the rental value of space occupied by the system. Analysis may be complicated further by variations in efficiencies of filtration with changes in air face velocity, although tests conducted [1] over a range of practical air velocities indicated little variation in efficiency when determined on a weight basis.

VELOCITY-RESISTANCE RELATIONSHIP

One of the important variables affecting the economy of air filtration is the relationship between air velocity and filter pressure drop. Theoretically, if the air flow through the filter is completely turbulent, the resistance to flow varies as the square of the velocity, while if the flow is laminar or stream-

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[1] Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity through Filter, by F. B. Rowley and R. C. Jordan. (ASHVE TRANSACTIONS, Vol. 44, 1938.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, San Francisco, Calif., June, 1941.

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lined, the resistance varies directly as the first power of the velocity. Because of the nature of the media through which the air is passing, it would appear likely that in the case of mechanical air filters a combination of both laminar and turbulent flow might occur.

In order to determine the actual relationship between air velocity and filtration energy losses, four typical filters were chosen for experimental laboratory tests. These filters may be described briefly as follows:

A—A permanent type of cleanable oil filter, 4 in. thick, with 24 layers of expanded metal and wire screen graded from coarse mesh at entrance to fine mesh at leaving side.

B—A viscous coated throw-away type filter 2 in. thick. The fibrous media were graded in fiber size, density, and oiling from entering to leaving side.

C—A cellular type filter 2 in. thick built in 2 sections, each with the axis of cells set at 45 deg to the center line of duct and at 90 deg to each other. The cells on the entering side were of larger dimensions than those on the leaving side of the filter.

D—A filter of cotton media of coarse material on the entering side and glazed on the leaving side. The filter media were accordion plaited in frame to give an area of approximately 12 times the cross sectional area of the air stream.

Each of these filters was subjected to face air velocities of 200, 300, 400 and 500 fpm under conditions of no dust loading and at dust loadings corresponding to approximately 0.2, 0.3, 0.4, and 0.5 in. of water filter pressure drop at a 300 fpm face air velocity. These velocity-resistance tests were made at various conditions of dust loading in order to determine what changes, if any, were effected by differences in the dust loading pattern. Because of the accordion plaited arrangement of the filtering media in Filter *D*, these face air velocities (as based upon the projected filter area) were equivalent to media face velocities of only 16, 24, 33, and 41 fpm. The filter test apparatus used for determining the air velocities and the filter energy losses, and for feeding the dust to the filters has been described in previous publications.[2, 3] The dust used in loading the filters was fed at a rate of 40 grams per hour and consisted of 50 per cent by weight of Pocahontas ash, 20 per cent Illinois fly ash, and 10 per cent Fuller's earth, all screened through a 200 mesh screen, and 20 per cent lamp-black screened through a 100 mesh screen.

As a result of these tests it was found that the relationship between velocity and filter resistance could be expressed most accurately by the equation,

$$R = KV^n + C \quad . \quad . \quad .

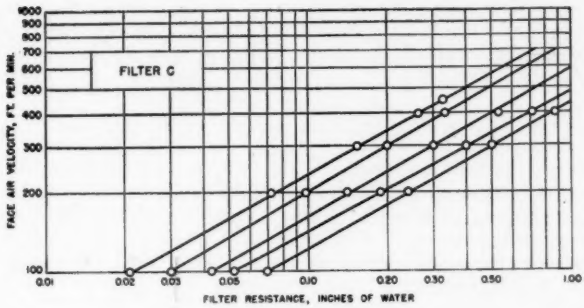

FIG. 1. VELOCITY-RESISTANCE CURVES FOR FILTER C UNDER DIFFERENT CONDITIONS OF DUST LOADING

Filters *C* and *D*. Table 1 shows a summary of the velocity-resistance relationships for all filters tested as obtained from these and similar graphs. The exponent, n, is determined from the slope of the graph; the constant, K, varies with both the velocity and the dust loading and has not been recorded as it has little practical significance. The closest approach to completely turbulent flow is found with Filter *C*, in which the resistance varies as the 1.83 power of the velocity. The closest approach to laminar flow is found with Filter *D*, in which the resistance of the filter, other than for the initial unloaded condition, varies as the 1.21 power of the velocity. It is interesting to note that with this filter the velocity-resistance relationship is different for the initial unloaded condition than it is when containing some dust load. Apparently the flow is more nearly turbulent when unloaded than when loaded. All of these tests indicate that for unit mechanical filters of usual design the air flow can be classed as neither completely laminar nor completely turbulent, but probably a combination of the two. It is probable that the conditions of flow change in different sections of the filters because of the gradients in media size and

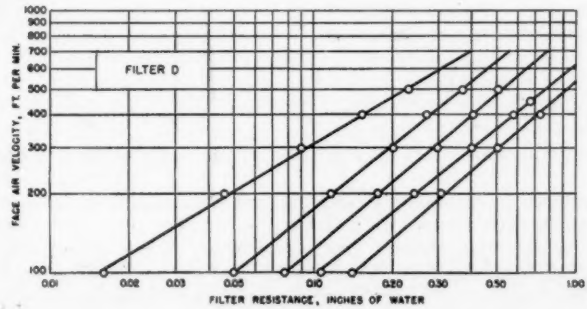

FIG. 2 VELOCITY-RESISTANCE CURVES FOR FILTER *D* UNDER DIFFERENT CONDITIONS OF DUST LOADING

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pack density. The range of velocities used in these tests was limited to approximately that encountered in practice and it is probable that if this range had been extended, marked variations in the velocity exponents would have been found.

EFFECT OF VELOCITY ON FILTER LIFE AND DUST HOLDING CAPACITY

In order to illustrate the effect of face air velocity on filter life and dust holding capacity a series of standard performance tests was run on all four filters by the weight or crucible method [4] using the 50-20-20-10 dust mixture mentioned earlier at a rate of dust feed of 40 grams per hour and a face air velocity of 300 fpm. The resistance curves thus obtained, along with the

TABLE 1—VELOCITY-RESISTANCE RELATIONSHIPS FOR FILTERS *A*, *B*, *C*, AND *D*

FILTER	CONDITION OF FILTER	VELOCITY-RESISTANCE RELATIONSHIP
A	Both unloaded and with all dust loads	R varies as $V^{1.72}$
B	Both unloaded and with all dust loads	R varies as $V^{1.60}$
C	Both unloaded and with all dust loads	R varies as $V^{1.83}$
D	Unloaded	R varies as $V^{1.67}$
D	With all dust loads	R varies as $V^{1.21}$

relationships of Table 1, were used to determine the resistance curves for face air velocities of 200, 400 and 500 fpm. Figs. 3 and 4 show typical families of resistance curves obtained for Filters *C* and *D* in this manner.

These resistance curves may be used as a basis for determining the effect of face air velocity on filter life and dust holding capacity if an arbitrary limiting resistance is set beyond which it may be considered uneconomical to operate the filter. This is difficult to do, however, as such a limiting resistance depends upon a variety of conditions. With the majority of installations 0.4 in. of water is usually considered as a reasonable limit with a face air velocity of 300 fpm, but this does not necessarily apply to other velocities. In general, after a sufficiently heavy deposit of dust has been collected in the filter, the resistance curve tends to rise so rapidly that increasing the allowable maximum resistance has comparatively little effect in lengthening the life and dust holding capacity.

Two methods of determining filter life and dust holding capacity were used in the present calculations. In one case the limiting resistance was taken as 0.4 in. of water, regardless of face air velocity; in the second, the life was determined by limiting the total resistance rise across the filter to 0.3 in. of water. The results of these calculations are shown in the first seven columns

[4] ASHVE Standard Code for Testing and Rating Air Cleaning Devices used in General Ventilation Work. ASHVE TRANSACTIONS, Vol. 39, 1933.)

of Table 2. In interpreting these results it must be remembered that the rate of feed was kept constant regardless of changes in air velocity, whereas in actual filtering installations increasing the face air velocity results in a proportional increase in the amount of dust fed to the filter. The results of

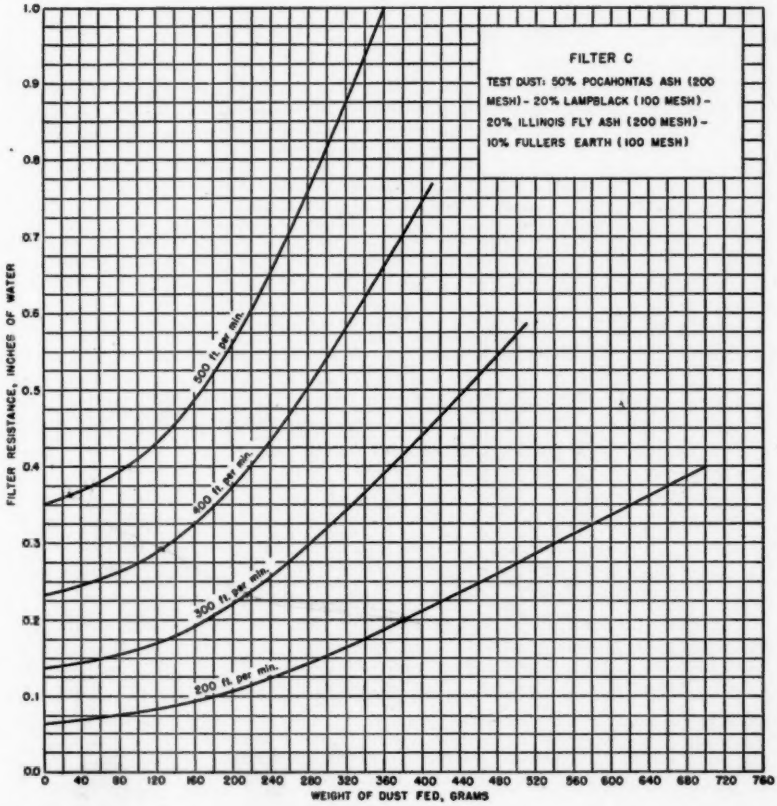
FILTER C
TEST DUST: 50% POCAHONTAS ASH (200 MESH) - 20% LAMPBLACK (100 MESH) - 20% ILLINOIS FLY ASH (200 MESH) - 10% FULLERS EARTH (100 MESH)
500 ft. per min.
400 ft. per min.
300 ft. per min.
200 ft. per min.
FILTER RESISTANCE, INCHES OF WATER
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
0 40 80 120 160 200 240 280 320 360 400 440 480 520 560 600 640 680 720 760
WEIGHT OF DUST FED, GRAMS

FIG. 3. RESISTANCE CURVES FOR FILTER *C* WITH DIFFERENT FACE AIR VELOCITIES (40 g per hour rate of dust feed)

these tests indicate a marked increase in filter life and dust holding capacity with decrease in air velocity. As would be expected, the greatest increases in performance are found in all cases when the limiting resistance is taken as 0.4 in. of water rather than when based on a 0.3 in. of water resistance rise. This is caused by the tendency for the initial filter resistance to approach and eventually to equal the limiting resistance as the velocity increases. This face

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velocity at which the limiting and initial resistances are equal, and consequently the life zero, is 436 fpm for Filter *A*, 575 fpm for Filter *B*, 537 fpm for Filter *C* and 1028 fpm for Filter *D*.

The ratios of the life in hours at 200 fpm to the life at 500 fpm when based upon a 0.3 in. of water resistance rise for Filters *A*, *B*, *C*, and *D* respectively,

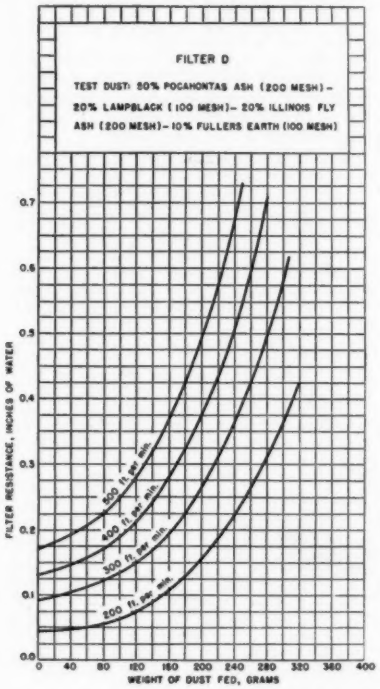
FILTER D
TEST DUST: 80% POCAHONTAS ASH (200 MESH) - 20% LAMPBLACK (100 MESH) - 20% ILLINOIS FLY ASH (200 MESH) - 10% FULLERS EARTH (100 MESH)
FILTER RESISTANCE, INCHES OF WATER
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
500 ft. per min.
400 ft. per min.
300 ft. per min.
200 ft. per min.
0 40 80 120 160 200 240 280 320 360 400
WEIGHT OF DUST FED, GRAMS

FIG. 4. RESISTANCE CURVES FOR FILTER *D* WITH DIFFERENT FACE AIR VELOCITIES (40 g per hour rate of dust feed)

are 1.4, 2.7, 2.7, and 1.5. The same ratios when based upon a final resistance of 0.4 in. of water are in the same order, infinity, 5.7, 8.0, and 1.8. This last set of ratios is somewhat misleading as it is unlikely that a 0.4 in. of water final resistance would be chosen in combination with a 500 fpm air velocity. However, a 400 fpm face air velocity in combination with this limiting resistance is quite possible, and certainly the first set of ratios based on a 0.3 in. of water resistance rise for all velocities and filters is a fair basis for comparison.

In order to obtain a rational basis for translating these laboratory values of filter life to actual filter life values a series of five tests was run to deter-

mine the normal weight concentrations of dust contained in air. These determinations were made by drawing samples of laboratory air through alundum crucibles and measuring the increases in crucible weight. The rate of sampling was 1 cfm and the time of sampling 300 min. The air-dust concentrations as determined by these tests ranged from 0.1175 to 0.0664 grains per 1,000 cu ft with an average value of 0.0845. Approximately this same average concentration was obtained as the result of an additional series of tests in which the concentrations were determined by electrical precipitation of the dust onto aluminum foil surfaces. All of these values were determined during periods when there were no unusual activities occurring in the laboratory which might cause abnormal dust concentrations and may be considered as representative of the concentrations to be found in typical urban areas.

The last two columns of Table 2 show the actual filter life in hours, based upon an air-dust concentration of 0.0845 grains per 1,000 cu ft and for both methods of determining limiting filter resistance. These values have been calculated by multiplying the laboratory filter life in hours by the ratio of the laboratory rate of dust feed to the rate of dust feed resulting from the passage of normal air through the filter. This latter rate of feed varies with the air velocity as it is assumed that in this case the weight of dust per unit volume of air remains constant. Therefore, it should be noted that, in order to remove the same amount of dust with an air velocity of 200 fpm as in the case of a 500 fpm air velocity, it is necessary to have 2½ times as much filtering area, as the rate of filtration is only 40 per cent as great. Whereas these values for actual filter life may be considered as reasonable approximations to what may be expected in practice, it should be remembered that values of both laboratory and actual filter life and dust holding capacity will vary widely with the nature of the test dust, and in these determinations it has been assumed that both the artificial laboratory test dust and the normal atmospheric dust suspensions have approximately the same physical characteristics.

Table 3 shows the variation of average filter resistance with face air velocity when based upon both a final limiting resistance of 0.4 in. of water and also a limiting resistance rise of 0.3 in. of water. These average resistance values were determined by finding the areas under the resistance curves (see Figs. 3 and 4) by means of a planimeter and then dividing these values by the abscissae. In all cases the average resistance increases rapidly with increased air velocity, and in general the average resistance at a 500 fpm face air velocity is approximately two to three times that at a 200 fpm face velocity. The air horsepower required to overcome the filter resistance is directly proportional to the average resistance, and therefore, power costs resulting from the use of high air velocities may become an important item.

DERIVATION OF FILTRATION COST EQUATION

It is apparent from the preceding analysis that the correct choice of face air velocity is important for economical filtration because of its effect on filter life and average resistance. However, in order to determine the most economical air velocity, it is necessary to consider concomitantly all cost factors which are affected by changes in air velocity. With this end in view, let

TABLE 2—EFFECT OF AIR VELOCITY ON LIFE AND DUST HOLDING CAPACITY FOR FILTERS *A*, *B*, *C*, AND *D*
(*Laboratory Test Dust Feed* 40 *g per hour. Dust mixture* 50, 20, 20, 10)

FILTER	AVERAGE EFFICIENCY, PER CENT	FACE AIR VELOCITY, FT/MIN	FILTER LIFE BY LABORATORY TEST, HOURS		DUST HOLDING CAPACITY BY LABORATORY TEST, GRAMS		RATE OF DUST FEED TO FILTER WITH AIR CONTAINING 0.0845 GRAINS PER 1000 CU FT, GRAMS PER HOUR	FILTER LIFE WITH AIR CONTAINING 0.0845 GRAINS DUST PER 1000 CU FT, HOURS	
			0.4 in. H_2O Final Resistance	0.3 in. H_2O Resistance Rise	0.4 in. H_2O Final Resistance	0.3 in. H_2O Resistance Rise		0.4 in. H_2O Final Resistance	0.3 in. H_2O Resistance Rise
A	80.3	200	19.0	19.0	610	610	0.183	4150	4150
		300	15.3	16.8	491	540	0.274	2235	2455
		400	7.0	15.2	225	488	0.366	765	1660
		500	0.0	13.8	0	444	0.457	0	1210
B	81.7	200	17.2	16.1	563	526	0.183	3750	3510
		300	9.2	10.1	301	330	0.274	1340	1470
		400	5.5	7.5	180	245	0.366	602	820
		500	3.0	6.1	98	199	0.457	263	534
C	69.9	200	17.5	16.2	490	454	0.183	3830	3540
		300	9.2	9.9	257	277	0.274	1350	1450
		400	5.5	7.4	154	207	0.366	601	809
		500	2.2	6.0	62	168	0.457	193	529
D	93.5	200	7.8	7.3	292	273	0.183	1710	1600
		300	6.3	6.2	236	232	0.274	921	905
		400	5.2	5.5	195	206	0.366	570	603
		500	4.3	4.9	161	183	0.457	378	430

TABLE 3—EFFECT OF AIR VELOCITY ON AVERAGE RESISTANCE FOR FILTERS *A*, *B*, *C*, AND *D*

(*Dust Feed* 40 *grams per hour. Dust mixture* 50, 20, 20, 10)

FILTER	AVERAGE RESISTANCE, IN. OF WATER (0.4 IN. WATER FINAL RESISTANCE)				AVERAGE RESISTANCE, IN. OF WATER (0.3 IN. WATER LIMITING RESISTANCE RISE)			
	Face Air Velocity, ft/min				Face Air Velocity, ft/min			
	200	300	400	500	200	300	400	500
A	0.15	0.26	0.36	...	0.16	0.28	0.43	0.61
B	0.20	0.23	0.28	0.35	0.20	0.25	0.34	0.43
C	0.20	0.23	0.29	0.36	0.18	0.25	0.35	0.47
D	0.14	0.18	0.22	0.25	0.13	0.18	0.23	0.27

Q = rate of filtering, cfm

t = time of filtering, hours per year

C = initial cost each filter (including installation labor charges) or cost of reconditioning, dollars

V = filter face air velocity, feet per minute

L = life of filter, hours

R_a = average resistance of filter, inches of water

H = average air horsepower[5] required for filtration

[5] See any treatise on fan engineering for derivation.

$$= \frac{R_a Q}{6350}$$

D = power cost, dollars per air horsepower-hour

A = filter area, square feet

I = total annual fixed charges or depreciation, per cent

B = cost original filtering installation, dollars per square foot filter area

F = rental value of space occupied by filters, dollars per square foot filter area per year

then

Volume of air to be filtered per year, cubic feet $= 60Qt$

Volume of air handled by each filter, cubic feet $= 60AVL$

Number filters (new or reconditioned) required per year $= \frac{Qt}{AVL}$

Cost of filters (new or reconditioned), dollars per year $= \frac{Qtc}{AVL}$

Power cost, dollars per year

$$= HtD$$

$$= \frac{R_a QtD}{6350}$$

Fixed charges, dollars per year

$$= \frac{QBI}{100V}$$

Rental value of occupied space, dollars per year $= \frac{QF}{V}$

The total cost of air filtration per year will then be the sum of these individual cost items, or

$$\text{Yearly cost air filtration} = \frac{QtC}{AVL} + \frac{R_a QtD}{6350} + \frac{QBI}{100V} + \frac{QF}{V} \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

In analyzing Equation (3) it is apparent that both R_a, the average resistance to air flow, and L, the filter life, are variables dependent upon air velocity. In a study of filtration cost in any specific instance, it is, therefore, necessary to evaluate these in terms of V, the air velocity.

In practically all instances, filter life, L, is an inverse function of V raised to some power greater than 1. Therefore, despite the fact that V appears in the denominator, the expression for cost of new or reconditioned filters per year generally increases as the velocity increases. As the average resistance, R_a, varies directly as some power of V, the yearly power costs will also increase as V increases. The yearly depreciation and space charges both decrease with increased velocity as both of these expressions contain V only in the denominator.

In determining filtration costs under practical operating conditions by means of Equation (3), there is in every case an optimum air velocity which results in minimum yearly filtration charges. This velocity may be determined by differentiating the yearly cost expression with respect to V, setting the resulting expression equal to zero and solving for V. However, the solution is, in most cases, quite difficult because of the number of terms in the original expression, and it is, therefore, usually much simpler to assume four or five velocities, determine the corresponding yearly filtration charges, and then determine the velocity corresponding to minimum cost graphically.

It must be realized that analytical methods such as are made possible by the application of Equation (3) have definite practical limitations because of the difficulties encountered in the evaluation of some of the terms. Usually an approximate numerical expression must suffice. However, calculation of the optimum velocity for each case encountered is not always necessary, as once the weight to be placed on each of the factors has been established by the analysis of a few cases, general engineering judgment will usually dictate velocities reasonably close to the optimum. The following analysis based upon actual data for a typical installation is presented as an illustration of the procedure to be followed in determining the optimum air filtration velocity.

Example: It is desired to determine the optimum filter face air velocity for an air filtration system to be operated in conjunction with the ventilation system of a building designed to handle 10,000 cfm for an average of 3,000 hours of operation each year. Unit, disposable filters, each 20 in. by 20 in., are to be used. The replacement cost of these filters, including labor charges, is $1.25 each. Power is available at $0.02 per kilowatt hour; or, assuming a fan efficiency of 70 per cent and a motor efficiency of 90 per cent, the power costs are $0.024 per air horsepower-hour. The total annual fixed charges on the installation, including interest charges on the capital expenditure and annual depreciation, are to be taken at 12 per cent.

In order to evaluate the yearly fixed charges and the rental value of the occupied space it is first necessary to determine the relationship between the number of filters and the cost of the necessary duct work and filter frames for the installation and the relationship between the number of filters and the floor

area necessary for the installation. In the solution of the present problem the duct work for several sizes of installations was drawn up, estimates were obtained on the cost of this duct work, and the floor area required for these installations measured. These relationships were thus found to be roughly linear. For example, in the case of the initial installation cost, it was found that in taking the number of filters in the installation at 9, 18, 27, and 36 the minimum cost per square foot of filter area was \$3.71 and the maximum \$4.32. Therefore, the initial cost of duct work and filter frames may be taken as approximately \$4.00 per square foot of filter face area. In the same way the rental value of the space occupied by the additional equipment (duct work,

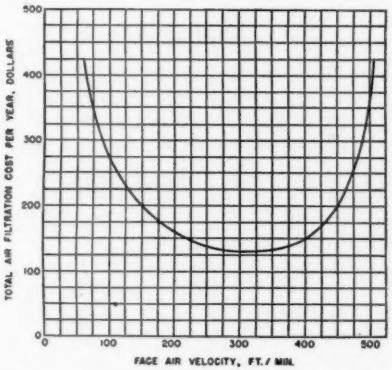

FIG. 5. GRAPHICAL SOLUTION OF EXAMPLE SHOWING VARIATION OF YEARLY FILTRATION COSTS WITH FACE AIR VELOCITY. RATE OF FILTRATION 10,000 CFM

filter frames, etc.) required for the filtration system was determined and may be taken as approximately \$1.90 per year per square foot of filter face area. These figures are based upon the prevailing costs for the particular locality under consideration.

In order to evaluate the power costs and the cost of replacement filters it is first necessary to determine the relationship between actual life in hours, L, (based on a limiting resistance of 0.4 in. of water and in dust concentration 0.0845 grains per 1000 cu ft air) and face air velocity, V, and the relationship between average resistance, R_a, and air velocity, V. Actual laboratory tests (similar to those shown in Table 2) were made on the particular filter chosen for installation and these relationships found to be

$$L = \frac{(600 - V)^{3.64}}{1972}$$

and

$$R_a = \frac{V^{0.704}}{227}$$

In the solution of this problem, face air velocities of 100, 200, 300, 400, and 500 fpm were chosen and the yearly filtration costs calculated by substitu-

tion of the given conditions in Equation (3). The graphical solution, with these air face velocities plotted against the calculated total yearly filtration costs, is shown in Fig. 5. This curve indicates that the optimum air velocity, as far as filtration costs are concerned, is in the neighborhood of 300 fpm. In this particular case, any air velocity chosen between 200 and 400 fpm would be reasonably satisfactory. However, below 200 fpm and above 400 fpm, the air filtration costs rise very rapidly and at 100 fpm are approximately double and at 500 fpm almost triple the minimum cost. If filter life had been determined for a limiting resistance rise of 0.3 in. of water instead of the basis used, the yearly filtration costs would not have increased quite as rapidly with the higher air velocities.

SUMMARY

In this paper it has been shown that the relationship between filter resistance, R, and the face air velocity, V, for mechanical filters is given by the equation $R = KV^n$ where K and n are constants and V is limited to practical operating velocities. The exponent n is dependent upon the design of the filter and in some cases upon the filter loading. For all filters tested the values of n were between 1 and 2 indicating that the air flow through the filters was neither completely laminar nor completely turbulent, but probably a combination of the two.

It has been shown that the face air velocity chosen in the operation of a filter has a marked effect upon the life and dust holding capacity of the filter when these performance characteristics are based on either a final limiting resistance or upon an allowable resistance rise. When the laboratory values of filter life are translated into actual values of filter life, the effect of air velocity is even more marked, since under practical operating conditions decreasing the air velocity decreases also the rate of dust feed to the filter.

The total yearly cost of air filtration has been shown to be dependent upon several factors each of which varies to some degree with the face air velocity. An equation has been derived showing the relationship between air filtration costs and air velocity. The procedure has been outlined by which it is possible to determine the optimum air velocity for minimum filtration costs.

No. 1188

LOCAL COOLING OF WORKERS IN HOT INDUSTRY

By F. C. HOUGHTEN,* D.Sc., M. B. FERDERBER,** M.D., AND CARL GUTBERLET,†
PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

THIS paper gives the results of studies involving physiological reactions of persons to hot atmospheres which were carried on at the ASHVE Research Laboratory, U. S. Bureau of Mines, Pittsburgh, Pa., under the Technical Advisory Committee on Air Conditioning in Industry, consisting of: W. L. Fleisher, *Chairman;* Philip Drinker, Dr. Leonard Greenburg, H. P. Greenwald, A. M. Kinney, J. W. Kreuttner, L. L. Lewis, H. B. Matzen, Dr. W. J. McConnell, Dr. C. P. McCord, P. A. McKittrick, Dr. R. R. Sayers, Charles Sheard, C. Tasker, R. M. Watt, Jr., and H. E. Ziel. Previous reports on the subject are listed in the bibliography.

The usual conception of air conditioning for human comfort assumes its application for the purpose of providing ideal comfort. A field offering a challenge to the air conditioning engineer is rapidly developing in which air conditioning is applied to alleviate undue discomfort and injury to the health of workers in hot industries. This has sometimes been referred to as air conditioning in industry. In either case, the atmospheric environment is controlled in order to gain the desired effect on the physiological responses including the feeling of comfort or discomfort. In one case the person benefited by air conditioning expects to enjoy the optimum condition for a comfortable existence; while in the other case it is frequently assumed that the best that can be afforded is an environment which will alleviate great distress, and avoid violent physiological derangement which may either cause excessive current suffering or permanent injury to the health of the worker, reduce his output, or cause him to avoid employment at the work in question.

The relations between the physical activity of the person and his rate of heat production on the metabolism on the one hand, and the rate of dissipation of this heat to the atmospheric environment on the other hand have been amply established for a wide range of activities and atmospheric conditions through research by the ASHVE Laboratory in cooperation with the U. S. Bureau of Mines, the U. S. Public Health Service, and others. These studies

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, San Francisco, Calif., June, 1941.

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show that the rate of heat production ranges from 53 cal per square meter per hour when seated at rest to as high as 180 cal per square meter per hour at very hard labor or 118 cal per square meter per hour for labor construed to be as hard as the average worker cares to be engaged in continuously for two 4-hour work periods per day. For a person at rest this heat may be dissipated with ideal comfort of from 66 to 72 deg ET. Perspiration occurs at about 76 deg ET and a rise in body temperature and other physiological derangement begins to occur when the temperature reaches about 82 deg ET. For a person engaged in continuous labor with a metabolic rate of 118 cal per square meter per hour a lower effective temperature is required for ideal comfort, and a rise in body temperature and other physiological derangement occurs at about 75 deg ET. The relation between rise in body temperature and the environmental conditions for a person at light work taken from an earlier laboratory study [1] are shown in Fig. 1. It is obvious from these facts that in order to avoid violent physiological derangement alleviation is necessary in many industries if men are to work regular shifts. The air conditioning engineer is naturally called upon to supply improved conditions through air conditioning.

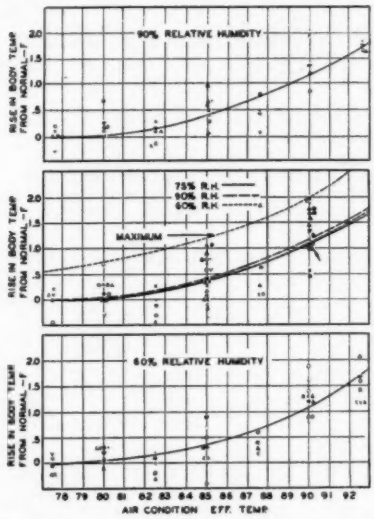

FIG. 1. RELATION BETWEEN RISE IN BODY TEMPERATURE FROM NORMAL DURING 3-HOUR EXPOSURE AND THE EFFECTIVE TEMPERATURE OF THE AIR CONDITION IN THE TEST ROOM FOR 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES. TEST POINTS FOR INDIVIDUAL SUBJECTS APPLY TO HEAVY SOLID CURVES

Where the occupancy density in a hot industry is high, the most logical, as well as the most feasible, course is to air condition the entire environment so as to supply ambient air in the entire enclosure sufficiently satisfactory to give the desired results. There are many instances, however, where one or at most a small number of men are engaged in a large enclosure in which an industrial process requiring, or resulting in, hot surroundings is located. For such conditions the conditioning of the entire space is highly uneconomical or even impossible from a practical point of view. Here, the application of local cooling to take care of the few occupants without conditioning the entire ambient surroundings may be resorted to by use of one of the following methods: (1) In some instances the worker may be placed in a conditioned booth from which he may observe operations and perform his required tasks. (2) The worker may be supplied with a blast of cool, high velocity air into which he may step, thus providing himself with the degree of cooling required. (3) Cooled air may be blown through a loose-fitting, relatively air-

[1] Numbers refer to Bibliography.

tight suit so as to cool the worker without attempting to cool the surroundings. This paper includes the results of a study of the second and third methods of local cooling applications, the results being limited to atmospheric conditions of approximately 50 per cent relative humidity and either a 2500 fpm blast of cool air from a 4-in. nozzle located close to the worker, or a cool air supply of approximately 20 cfm through a loose-fitting *coverall* suit.

The study was made in the psychrometric rooms of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in the U. S. Bureau of Mines Building, Pittsburgh, Pa. Two subjects, one referred to as the *cooled worker* and the other as the *control worker*, spent the test period in one of the two psychrometric rooms in which the desired ambient air condition at approximately 50 per cent relative humidity was maintained. The cooled air was provided by a small forge type blower from the second psychrometric room so conditioned as to give cooling air of the desired dry-bulb and effective temperature with approximately 50 per cent relative humidity at the location of the worker.

In the blast study, the cooled worker spent most of his time either standing or sitting at rest in the blast of cool air, orienting himself at will to obtain the most desired effect. However, during approximately 1 min out of each 5-min period he left the cool blast to pull the work machine 10 times. This resulted in his being at ease in the cool blast approximately 80 per cent of the time and at work in the ambient condition the remaining 20 per cent of the time. The control worker divided his time between *at work* and *at ease* in a like manner. However, he was at no time permitted to enter the blast of cool air.

In the ventilated suit study, air of the desired dry-bulb and effective temperatures at approximately 50 per cent relative humidity was continually circulated through the suit of the cooled worker. The temperature of the ventilating air was observed in the 2-in. hose near the end where it was connected to the back of his *coverall* suit. During each 5-min period he was required to pull the work machine (Fig. 2) 10 times, which required about 1 min; the remaining approximately 80 per cent of the time he was allowed to sit or stand at ease. The control worker followed the same procedure with the exception that he was not supplied with cooling air through his suit.

The subjects were healthy young men from 18 to 22 years of age and of average size (approximately 19.5 sq ft body surface area) who were able to satisfactorily pass a physical examination given by a physician with industrial experience and classified by him as being good risks for workers in hot industries. In all, six different men served as subjects, two in each test. On succeeding days when the same subjects participated they alternated their positions as *cooled* and *controlled* workers. The clothing worn by all subjects in all tests included light weight underwear, low shoes, light weight socks, and coveralls.

The subjects entered the test room after resting 15 min in a relatively comfortable room. Just before entering the test room their body temperature, pulse rate, general feeling of warmth, and other physiological data were recorded. The collection of these data was continued through the period spent in the test room. Attention was particularly paid to the rise in body temperature, increase in pulse rate, degree of perspiration, and the comfort or discomfort experienced by the worker. In a few of the tests the metabolism for

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this degree of activity was determined and found to be 82 cal per square meter per hour. The subjects continued in the test room for 2 hours unless their body temperature reached 101 deg or their pulse 140 beats per minute, which were accepted as the safe limits beyond which the men should not be exposed.

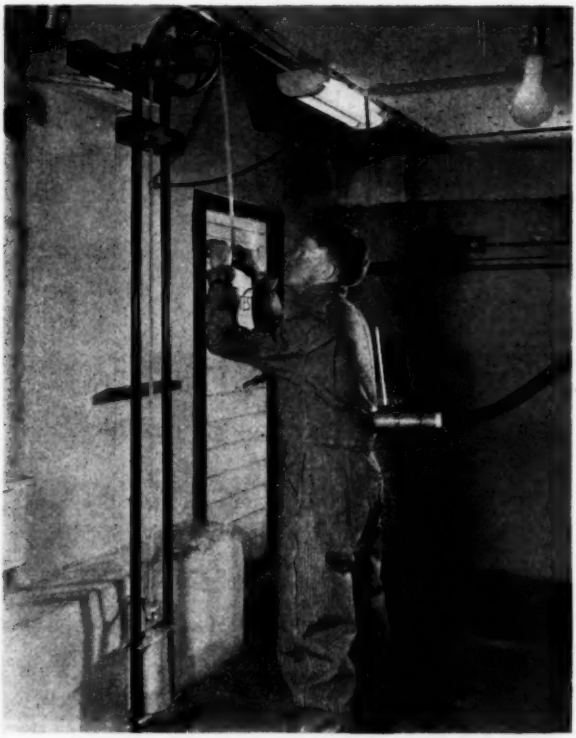

FIG. 2. WORKER IN VENTILATED SUIT OPERATING WORK MACHINE

When the reactions of either subject reached either of these limits he was immediately removed from the test which was however continued with the other subject.

TEST RESULTS

The rise in body temperature of the control worker, in both the blast and suit tests, was plotted against the dry-bulb temperature of the ambient condition at 50 per cent relative humidity in Fig. 3. The accompanying wet-bulb and effective temperatures of the ambient condition are indicated. For com-

parison, curves are included showing the relation between conditions of the atmosphere and rise in body temperature found in an earlier study for a person working at relatively hard labor,[2] a person working at very light work [1] moving about on his feet, and for a person seated at rest [3] all for still air conditions.

The relation between the rise in body temperature of the worker cooled by blast air at several dry-bulb temperatures at 50 per cent relative humidity and the ambient conditions are plotted in Fig. 4. Tests were actually made at, and the test points are plotted for, blast temperatures of 100 deg, 90 deg, 83 deg, and 75 deg dry-bulb. Heavy line curves for these points are drawn.

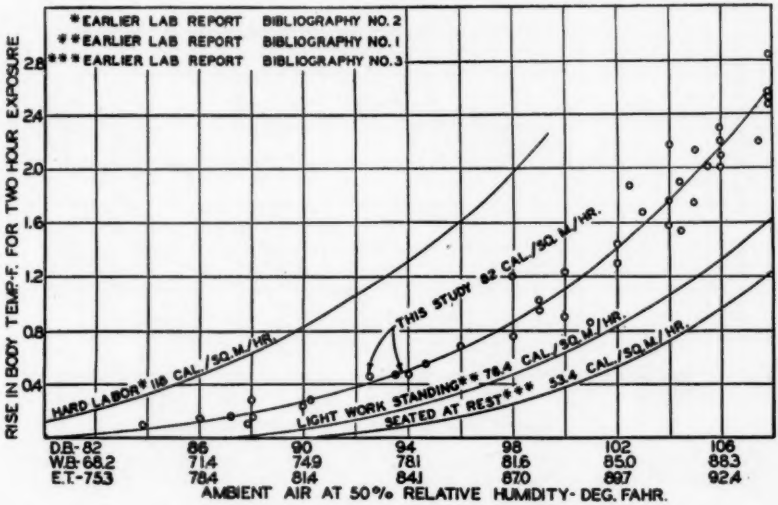
*EARLIER LAB REPORT BIBLIOGRAPHY NO. 2
**EARLIER LAB REPORT BIBLIOGRAPHY NO. 1
***EARLIER LAB REPORT BIBLIOGRAPHY NO. 3
RISE IN BODY TEMP-F. FOR TWO HOUR EXPOSURE
2.8
2.4
2.0
1.6
1.2
0.8
0.4
THIS STUDY 82 CAL./SQ.M./HR.
HARD LABOR* 118 CAL./SQ.M./HR.
LIGHT WORK STANDING** 78.4 CAL./SQ.M./HR.
SEATED AT REST*** 53.4 CAL./SQ.M./HR.
D.B.-82 86 90 94 98 102 106
W.B.-68.2 71.4 74.9 78.1 81.6 85.0 88.3
E.T.-75.3 78.4 81.4 84.1 87.0 89.7 92.4
AMBIENT AIR AT 50% RELATIVE HUMIDITY-DEG. FAHR.

FIG. 3. RELATION BETWEEN AMBIENT AIR AND RISE IN BODY TEMPERATURE OF CONTROL WORKER. CURVES FOR OTHER ACTIVITIES GIVEN FOR COMPARISON

These curves are expanded into a series giving the relationship between temperature rise of the worker and blast temperatures ranging from 100 deg to 75 deg dry-bulb at 50 per cent relative humidity.

The relation between the rise in body temperature of the worker cooled by ventilation of his suit is plotted against the dry-bulb temperature of the ambient condition at 50 per cent relative humidity in Fig. 5. Tests were actually made and the test points are plotted for ventilating air dry-bulb temperatures of 100, 90, 80, and 70 F, and these curves are expanded into a series giving the relation between rise in body temperature and ventilating air temperatures ranging from 100 to 70 deg dry-bulb at 50 per cent relative humidity.

The increase in pulse rate for the control and cooled workers was also observed and studied; however, they show approximately the same relation to rise in body temperature found in earlier studies.[1] Since rise in body temperature has been found to be the best and most consistent physiological change in

its correlation with the severity of the hot condition, only the rise in body temperature is here reported.

The control worker perspired profusely in the hot ambient conditions. While in the light work study,[1] Fig. 3, perspiration began to be noticed on the subjects at from 75 to 78 deg ET, the control subjects in this study engaged in the higher work output began to perspire sensibly at from 72 to 75 deg ET. Further, while for the light work studies perspiration did not soak completely through the clothing of the workers until an effective temperature of 88 or 90 deg was reached, this condition was observed for the control worker in this study at about 86 deg ET. The worker cooled by either the blast or the ventilated suit in this study was relatively free from perspiration

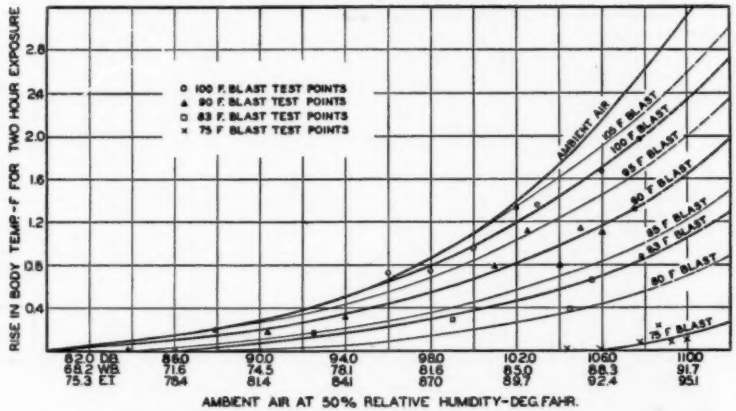

FIG. 4. RELATION BETWEEN AMBIENT AIR TEMPERATURE AND RISE IN BODY TEMPERATURE OF WORKER COOLED BY BLAST AIR AT 50 PER CENT RELATIVE HUMIDITY AND VARIOUS DRY-BULB TEMPERATURES

for all ambient conditions studied, when the cooling air temperature was as low as 80 deg dry-bulb with 50 per cent relative humidity. Until an ambient effective temperature of 95 deg, or a dry-bulb of 110 F at 50 per cent relative humidity was reached, 60 deg dry-bulb cooling air eliminated sensible perspiration entirely.

The data plotted in Figs. 3, 4, and 5 show that for the type of work engaged in by the control worker in this study he began to show a rise in body temperature when the dry-bulb temperature was 50 per cent relative humidity reached 80 F (or 74 deg ET) and he had a temperature rise of 1 deg in 2 hours when the dry-bulb temperature reached 99 deg. With ambient conditions of 80, 88, and 106 there was no rise in the body temperature of the cooled worker in blast temperatures of 90, 80, and 75 F, respectively. With the ventilated suit, slightly lower air temperatures were necessary. A few tests made with 60 deg air show that workers can perform the amount of work employed in this study for dry-bulb temperatures up to at least 112 deg with 50 per cent relative humidity.

Although the study was made with both ambient and cooling air temperatures at 50 per cent relative humidity, earlier work of the Laboratory showing that most physiological reactions, particularly rise in body temperature, are functions of effective temperatures, makes it possible to apply the results to other atmospheric conditions.

The study shows conclusively that workers may be kept free from body temperature rise, excessive increase in pulse rate, and from perspiration either with a blast of cool air or by ventilating their clothing with cool air. It should be emphasized, however, that variations in the manner in which the cooling air is handled or in the clothing worn may materially affect these results. Larger volumes of blast air or ventilating air through the suit

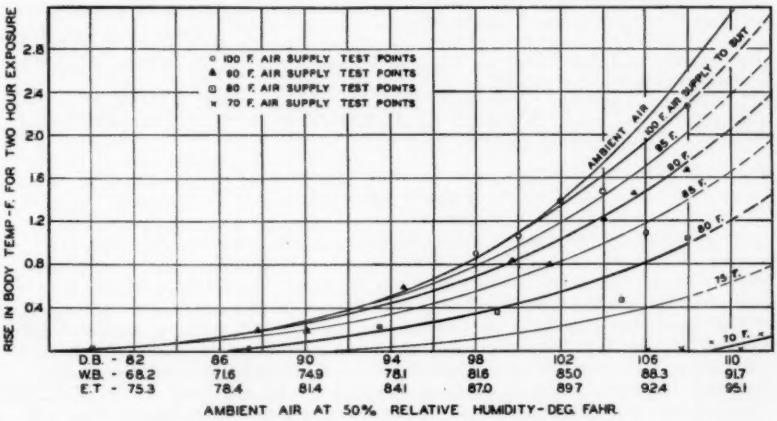
RISE IN BODY TEMP.-F. FOR TWO HOUR EXPOSURE
2.8
2.4
2.0
1.6
1.2
0.8
0.4
○ 100 F. AIR SUPPLY TEST POINTS
▲ 90 F. AIR SUPPLY TEST POINTS
□ 80 F. AIR SUPPLY TEST POINTS
× 70 F. AIR SUPPLY TEST POINTS
AMBIENT AIR
100 F. AIR SUPPLY TO SUIT
95 F.
90 F.
85 F.
80 F.
75 F.
70 F.
D.B. - 82 86 90 94 98 102 106 110
W.B. - 68.2 71.6 74.9 78.1 81.6 85.0 88.3 91.7
E.T - 75.3 78.4 81.4 84.1 87.0 89.7 92.4 95.1
AMBIENT AIR AT 50% RELATIVE HUMIDITY - DEG. FAHR.

FIG. 5. RELATION BETWEEN THE AMBIENT AIR TEMPERATURE AND RISE IN BODY TEMPERATURE OF WORKER COOLED IN VENTILATED SUIT, WITH VENTILATING AIR AT 50 PER CENT RELATIVE HUMIDITY AND VARIOUS DRY-BULB TEMPERATURES

would obviously give greater cooling effect and permit the worker to operate at higher ambient temperatures or in the same condition for longer periods of time. Likewise the same volume of air at a lower effective temperature will have the same effect.

The type of clothing worn is important. The cooling air will obviously have greater effect the closer it is kept to the body surface. In very hot conditions, particularly if radiant heat is an important factor, a given volume of cooling air blown through the suit will be more effective with greater insulation outside of the loose-fitting air tight suit; however, heavy clothing is objected to by the worker and should be avoided whenever possible.

Blast cooling air will be more effective the more intimately it comes in contact with the worker. In this connection it should be observed that after leaving the nozzle the cool air jet mixes rapidly with ambient air so that its temperature rises as the distance from the nozzle increases. The nozzle should be located near the worker. In this study it was found most effective to

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place the nozzle near the shoulder of the standing worker with the axis of the air stream projected downward at an angle so that the cooling air stream enveloped as large a portion of the body surface as possible. The worker was found, in general, to object to the blast of air over his head or face even though the sensation of coolness might be desirable.

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DISCUSSION

W. H. CARRIER (WRITTEN): I have read your recent papers on the effect of temperature and humidity conditions of people at work, especially at high temperatures. It occurs to me that we have overlooked one important thing in our study of effective temperatures, or to be more exact, the cooling power of air with people working.

The reason that a person does not respond to the wet-bulb temperature is that portions of his skin are dry and these portions are affected only by sensible heat convection and radiation. Those portions that are fully wetted are affected in direct accordance with the wet-bulb temperature as corresponding to a given air velocity, i.e., a temperature something higher than the theoretical wet-bulb temperature. This is the reason that a person that is fully bathed in perspiration, which is practically the limiting point at which he can dissipate the heat from his body, simulates very closely the conditions of a wet Kata thermometer. This would be true whether the perspiration was induced by his environment and normal metabolism at rest or at a much higher metabolism due to work. This critical point will, of course, occur at different wet-bulb temperatures depending upon the heat output. For example, with a person at rest, the wet-bulb temperature of the surrounding air, which might be the limit that it would take care of his metabolism, might be 93 deg in still air while under the same conditions working it might be as low as 90 deg or even lower. In both of these cases the slope of the effective temperature line would be practically the same and would be very close to the actual wet-bulb temperature. Where this temperature will occur will obviously depend upon the velocity of the air as well as the amount of work done, but the effect of velocity will be about the same at the lower temperature with a person working as it will at the higher temperature where the person is not working. Here again are changes in velocity effects upon the effective temperature.

What this amounts to is really a shift of scale in effective temperatures. This effect of shift in effective temperatures can be predicted approximately from known physical and physiological laws, together with the data available on people at rest, but it should be confirmed, with accuracy, by research data such as you are doing.

In looking over your data, it would seem that proper correlations have not yet been made and this obvious shift in effective temperatures has been overlooked. When

agreement of cooling power with wet-bulb temperatures is considered, it must be the wet-bulb corresponding to that in still air and not that with a 1,000 ft velocity which corresponds approximately to the temperature of adiabatic saturation. For example, at standard barometric pressure 110 deg and 106 grains of moisture per pound of air, the wet-bulb on the chart is 80 deg. In air at 15 fpm velocity, which is taken as approximately still air, the same wet-bulb temperature would be given at about 91 grains, if the curves given in "The Temperatures of Evaporation of Water Into Air," Carrier-Lindsay paper presented at the Annual Meeting of the Society in 1924, are taken as approximately correct. A line from 80 deg saturation to 91 grains and 110 deg will then be the approximate effective temperature line for a person whose body and clothing are completely saturated. If a person is working so that this condition is reached, i.e., he is working to the limit of his ability to dispose of heat at 80 deg saturation, then this line from 80 deg saturation to 110 deg and 91 grains will represent the effective temperature line and his limitation to dispose of heat will be at any point along this line with a 15 ft air velocity. This line is also the constant cooling line for a wet Kata thermometer at 15 fpm velocity. This suggests that you should use the cooling power of the wet Kata thermometer in discussing physiological reactions at temperatures at or above where there is a gain in body temperature, in other words, at or above physiological limitations of cooling. There will undoubtedly be some discrepancies in this relationship at lower humidities because of the failure of the body at some points of low humidity and high temperatures to furnish enough moisture to completely cover all parts of the skin or clothing. This effect will undoubtedly vary greatly with different individuals and also with their habits as to amount of liquids taken, etc. Probably the individual discrepancies will be greater than the average discrepancies from the theoretical relationship, hence the necessity of having a considerable number of subjects and repeating the same observations with the same subjects.

W. L. FLEISHER: I would like to say, being Chairman of this Committee, that this paper insofar as industry is concerned has great significance. It develops something very interesting in the present emergency, where the owner, manufacturer and the government itself are quite undetermined as to what conditions they should provide for the men so that they may properly manufacture the goods required. We have been working for the last few years to find the maximum conditions under which people can work and produce the goods economically in all of these so-called black-out air conditioned buildings. We find there is a great controversy as to what these maximum conditions should be, or whether refrigeration is required to produce the right conditions. Very definitely all of the work we have done this past year would indicate that so-called effective temperature of 78 deg as an average is good enough for the worker. At that point he feels no discomfort and can produce the amount of work required with no physiological changes. Nevertheless, quite a number of people working with government funds are calling for better conditions. It is my contention that if the owner were paying for the conditioning system and not the government, he wouldn't dream of providing any further aberration than a 78 deg ET.

The one question that comes up that Dr. Houghten might explain is the incident of sweating or perspiration. The companies contend that when working on polished machinery, conditions must be kept down to a point where the men's hands do not sweat or where sweat is not deposited on the goods.

MR. HOUGHTEN: In many hot ambient conditions the clothing of the worker not provided with local cooling became wet or even saturated with perspiration in a relatively short time and remained so as long as he remained in the condition. The worker provided with local cooling did not perspire noticeably.

The report indicates the effective temperature of the ambient condition at which various degrees of perspiration may be expected for both the control worker and for the worker cooled with air at different temperatures. The difference is very profound and indicates that perspiration of the cooled worker can be entirely controlled over a wide range of hot ambient conditions.

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S. H. DOWNS: Dr. Houghten intimated from the digest of the paper that the difference in the temperature required for the air in the suit and in the blast where the man is working was due entirely to the volume of air. Could you tell us from your investigations whether or not the difference in temperature of the air in breathing had some effect on that difference in the result?

MR. HOUGHTEN: When the worker was cooled with the blast a much greater volume of air was required than was the case when his suit was ventilated with cooled air. This was for the obvious reason that after leaving the nozzle the blast of cool air expands in cross-section and mixes rapidly with the ambient air. As a result its cooling power diminishes both due to a rise in the temperature of the mixture and a decrease in its velocity. As a result the cooling effect per unit volume of blast air was less efficient than was the case when the cool air was blown through the suit, when a larger percentage of the cooling air came more intimately in contact with the body surface of the worker. It has been demonstrated that any cooling applied to any part of the body serves to lower the entire body temperature. As a result you may cool a man working in a hot ambient condition by blowing air of sufficient velocity and of low enough temperature over any relatively small portion of his body surface. Cool air blown over the head will remove a certain amount of heat from the surface as well as provide cooling through the respiratory tract due to the cooler air taken in. However, earlier laboratory studies of heat and moisture dissipation from the bodies of persons indicate that the relatively small volume of air used in breathing is a small factor in the elimination of the entire amount of heat which must be dissipated from the body. Likewise the surface area of the head is not sufficiently great to remove enough heat to be a great factor in cooling the man. Hence, it was more effective to pass cooled air over the larger portion of the worker's body either in the form of the blast or through the clothing.

MR. DOWNS: The other viewpoint, that the effect of breathing cool air is psychological, wouldn't that affect your answer as to how comfortable they feel since you determine the effect of temperature by answers to questions of how they feel?

MR. HOUGHTEN: A person appears to offer a more immediate psychological response to cool air passing over the face and head, and it was apparent in the study that a hot worker if given the opportunity would first prefer to keep his head in the cool blast. This effect as pointed out previously is largely psychological and is not of great moment in the actual amount of heat removed from the body, which after all must be the determining factor in keeping down body temperature.

While the worker seemed to prefer cool air passing over his face for a short while if he was hot, he soon learned that he could keep down his body temperature and thus relieve his distress more effectively by having the air pass over the larger part of his body surface. Further, the high velocity blast of air became very annoying when continued over a long period of time and, as a result, during most of the test periods the workers avoided having the air blow over their faces. This was particularly true when the cooling was sufficient to keep down their body temperature. It should be emphasized that the distress of the worker in hot atmospheres is largely wrapped up with his elevation in body temperature and other physiological changes and not due to the sensation of hot air on either the whole or any part of his body surface including his face. While this may not be absolutely true it is true to a large extent. It is not as distressing to a person to breathe hot air when his physiological reactions are not deranged due to excessive heat as it is for him to experience these physiological derangements even though he has cool air to breathe and in contact with his face.

No. 1189

PERFORMANCE CHARACTERISTICS OF A FORCED DRAFT, COUNTERFLOW SPRAY COOLING TOWER

By Howard H. Niederman,* E. D. Howe,** John P. Longwell,*** Berkeley, Calif.
Ralph A. Seban,† Palo Alto, Calif., and L. M. K. Boelter,‡ Berkeley, Calif.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the University of California.

In small spray cooling towers, in contrast to packed cooling towers, the water distribution across any section varies widely. This distribution is a function of the type of spray used, the water pressure, water velocity, water temperature, air velocity and pressure, and of the tower construction. In general the water pressures and velocities for any given rate of flow are fixed by the type of spray, while the air velocities and pressures for any given rate of flow are fixed by the tower design and partially by the characteristics of the water spray.

There are many types of spray nozzles on the market today, but the manufacturers thereof do not have data available which will yield drop sizes and distributions. Without adequate data it is extremely difficult to predict what the performance of any given installation will be. In the present work several different types of sprays were used and their over-all characteristics noted.

Generally speaking the performance of cooling towers is expressed by means of the rate coefficients for thermal convection, mass transfer, and total energy transfer:

$$Gs(t_1 - t_2) = f_c a\ V(T_1 - t)_{lm} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

$$G(H_1 - H_2) = f_d a\ V(H_i - H)_{lm} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$G(h_1 - h_2) = f_h a\ V(h_i - h)_{lm} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Equations (1) and (2) are the integrated forms of the differential equations

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Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, San Francisco, Calif., June, 1941.

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which are derived in reference (16),[1] and which have been presented in the paper of London, Mason, and Boelter (8). Equation (3) has been derived in the same paper, and further information on the use of the equation has been given in the recent paper of London, Nottage, and Boelter (9). The unit volume

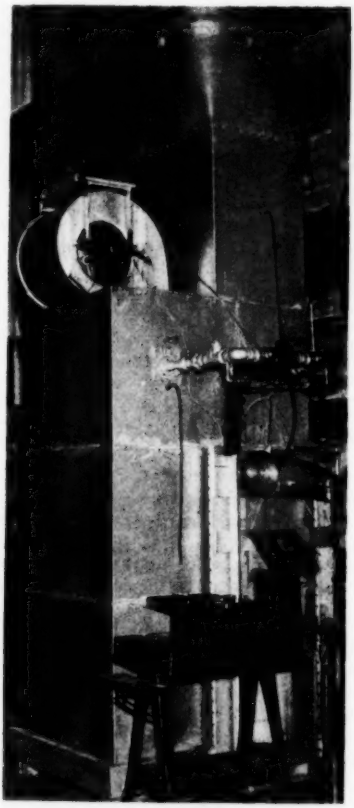

FIG. 1. VIEW OF SPRAY COOLING TOWER SHOWING THERMOCOUPLES, POTENTIOMETER, AND AIR MEASURING DEVICES

conductances, $f_c a$, $f_d a$, and $f_h a$ are products of an over-all conductance between the main body of water and the main body of air, and the transfer area per unit volume of the exchanger. The main body may be defined as that part of the substance existing at a temperature indicative of the energy content of all

[1] Numerals in parentheses refer to Bibliography.

of that substance. The use of logarithmic mean potentials in Equations (1), (2) and (3) is questionable. The logarithmic mean potential will yield the correct transfer rate only in those cases for which the operating line and the equilibrium line are straight over the region in question. It is doubtful whether this condition ever obtains in a spray tower, since flow characteristics change rapidly with height. The results, consequently, may only be compared accurately with the behavior of dynamically similar towers, the performances of which are calculated on the same basis. Any comparison with packed towers or wetted wall towers in which the logarithmic mean applies (under the conditions specified) can only be approximate. However, for want of a better, simply

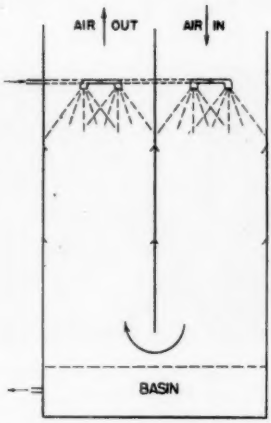

FIG. 2. FLOW DIAGRAM OF SPRAY COOLING TOWER
(4 ft x 3 ft x 7 ft)

computed potential the rate coefficients presented in this paper are calculated on the basis of the logarithmic mean.

In order to determine whether calculation of the conductances in terms of the number of transfer units (NTU) would yield a different result from that using the logarithmic mean potential, several computations were made. The conductances obtained in the former manner agreed within 2 per cent of those obtained by the latter method.

DESCRIPTION OF APPARATUS

The tests were made on a forced draft, spray cooling tower illustrated in Fig. 1 which was divided into two vertical sections, one section providing parallel flow and the other counterflow as shown in Fig. 2. Water circulation during runs took place in a closed system, the only loss being that due to evaporation. Water was pumped from the basin of the tower through a calibrated metering orifice and thence through a heat exchanger and back into

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the top of the tower where it was introduced through six sprays in each section. The rate of evaporation was obtained by noting the change in level of the basin with respect to time. Measurements of water level were made by means of a point gage shown in Fig. 3.

Air was forced through the tower by means of a fan placed at the entrance to the parallel flow section and was metered at the outlet. The necessary temperature measurements (*i.e.*, wet- and dry-bulb temperatures of the air in and out of the tower and the temperatures of the water in and out of the tower) were evaluated from thermocouple voltages, the thermocouples having been appropriately positioned at the entrance and exit tower sections.

The piping into the tower was so arranged that the parallel and counterflow sections might be run individually or simultaneously. Inasmuch as it is

FIG. 3. POINT GAGE, AND OUTLET FOR WATER DISTRIBUTION PAN

difficult to determine what the driving potentials are in a parallel-counterflow system such as this, it was decided to take data on the two sections individually. The results presented here are from data taken on the counterflow side.

The construction of the tower was such that the water after leaving the sprays struck the walls within a very short distance. The drops then either coalesced and traveled down the wall in sheets, or else were deflected back into the air stream. The cooling of the water was therefore obtained from both the drop surfaces and the surfaces exposed along the walls. Several protrusions are located circumferentially on the inside wall of the tower. These asperities cause mixing of the water sheet and separation from the wall.

Since the conductances are dependent upon the transfer area per unit volume, they are inevitably dependent upon the water distribution and drop size distribution. Any conditions which will cause the transfer area, a, to increase will consequently increase the over-all conductances. The extent to which transfer area depends on drop size distribution may be seen in Table 1 which gives the surface area exposed when a given volume of liquid is distributed into drops of various sizes according to several different methods of distribution.

The given liquid volume in the case under consideration was chosen as one

which might reasonably exist in 1 cu ft of the cooling tower at any moment if the water were evenly distributed throughout.

The most obvious method of obtaining greater transfer area in a spray tower is to attempt to obtain a spray distribution which will give a maximum number of fine drops throughout the length of the tower, but these drops must be large enough so that they will not evaporate throughout their trajectory in the tower. Several different types of sprays were employed for this purpose.

Until recently (8) the performance of cooling towers has been represented in terms of the over-all coefficient for mass transfer, $f_d a$. In order to facilitate direct comparison the results obtained are given in the same manner, although the conductances for sensible heat transfer and total energy transfer have been calculated and are available for comparison with new data as they are presented.

The values of conductances were computed from runs which were of sufficient duration to yield accurate measurements of the lowering of the basin level, and during which time steady state conditions held. The accuracy of the data obtained was determined from the energy and mass balances. Only those runs in which the balances checked within seven per cent were utilized.

ENERGY BALANCE

$$L\, c_p(T_1 - T_2) + \Delta L\, c_p\, (T_2 - 32) = G(h_1 - h_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The left-hand side of Equation (4) indicates the energy gain of the air. The datum for both water and for air is 32 F. The enthalpies of the air at entrance and exit were calculated by means of Equation (5).

$$h = s\,(t - 32) + 1061H \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The error of the energy balance was obtained by dividing the difference between the two sides of Equation (4) by the energy lost by the water. No correction was made for the thermal energy loss from the uninsulated sides of the tower. Measurements of the outside wall temperatures of the tower during several runs, taken by means of thermocouples, yielded values of the conductance (convection plus radiation) of approximately 0.8 Btu per foot^2 hour F (evaluated by means of the Heilman equation). Measurements of the over-all conductance for heat transfer through the walls to the outside air made by R. A. Seban (12) indicated a value of 1.6 Btu per foot^2 hour F. His method involved insulation of the ends of the tower and determination of the heat loss of the water through the tower with no air flowing. The conductance was then calculated directly from the equation.

$$q = f\, A\, \Delta T \quad . \quad (6)$$

The magnitude obtained by Seban, however, was at temperatures 30 to 40 F above the usual water temperatures within the tower. In either case, the thermal energy loss through the walls represents less than 2 per cent of the total energy transfer from the water to the air stream.

MATERIAL BALANCE

$$\Delta L = G\, \Delta H \quad . \quad (7)$$

The left-hand side of Equation (7) gives the rate of evaporation from the basin of the tower as measured by the change in point gage level, while the

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TABLE 1—CHANGE IN SURFACE AREA OF DROPS

(Assume that the water is distributed uniformly within the tower and that there is 0.004 ft³ water per ft³ of tower volume at any moment. Assume that the drops are all spherical and that there are only five drop sizes, ½, 1, 1½, 2, 2½ mm diameter)

DROP SIZE		DISTRIBUTION ACCORDING TO EQUAL VOLUMES			DISTRIBUTION ACCORDING TO EQUAL NUMBERS OF DROPS			PROBABILITY DISTRIBUTION OF VOLUME			PROBABILITY DISTRIBUTION OF NUMBER OF DROPS		
Diameter Mm.	Millifeet	Number of Drops	Volume x 10^6 ft³	Surface Area ft²	Number of Drops	Volume x 10^6 ft³	Surface Area ft²	Number of Drops	Volume x 10^6 ft³	Surface Area ft²	Number of Drops	Volume x 10^6 ft³	Surface Area ft²
0.5	1.64	347,000	800	2.93	7,720	17.8	0.06	121,800	280	1.02	3,340	7.7	0.03
1.0	3.28	43,400	800	1.46	7,720	142.2	0.26	52,100	960	1.76	11,450	211.5	0.39
1.5	4.92	12,800	800	0.98	7,720	481.0	0.59	24,300	1520	1.85	18,100	1131	1.38
2.0	6.56	5,420	800	0.73	7,720	1140	1.04	6,500	960	0.88	11,450	1687	1.54
2.5	8.20	2,780	800	0.59	7,720	2225	1.63	970	280	0.20	3,340	963	0.70
Totals.....		411,400	4000	6.69	38,600	4006	3.58	205,670	4000	5.71	47,680	4000	4.04

Area Per Unit

Volume of Tower = 6.69 ft²/ft³ 3.58 ft²/ft³ 5.71 ft²/ft³ 4.04 ft²/ft³

If the drops were all of the same average size, 1½ mm., the transfer area would be 'a' = 4.87 ft²/ft³.

right-hand side gives the change in moisture content of the air passing through the tower. The error was calculated by dividing the difference of the numerical magnitudes of the two sides of equation (7) by ΔL.

Since the water distribution within the tower is extremely important an attempt was made to determine what fraction of the water passed down the central portion of the tower. This was done by placing a collecting pan having an area equal to one-half the tower cross section in the center of the bottom of the counterflow side. The results are shown by the curves in Fig. 4. In Fig. 5 will be found a detailed description of each one of the five nozzles. It is apparent that the best space distribution was given by Nozzle *D*

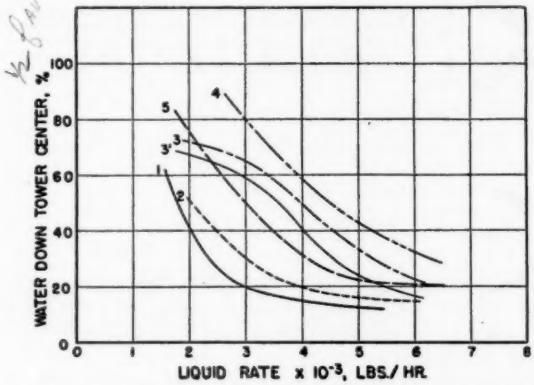


FIG. 4. WATER DISTRIBUTION WITHIN TOWER MEASURED AT BOTTOM OF TOWER

Per cent passing down tower center versus liquid rate

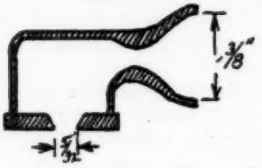
Legend:

1. Nozzle *A*, $G = O$
2. Nozzle *B* with cylindrical bore, $G = O$
3. Nozzle *C* with 1 in. nipple, $G = O$
3'. Nozzle *C* with 1 in. nipple, $G = 5000$ lb/hr
4. Nozzle *D*, $G = O$
5. Nozzle *E*, $G = O$

and the worst by Nozzle *A*. These results are misleading, however, for they give the water distribution in only one particular portion of the tower, *i.e.*, at the bottom, and they give no indication whatsoever of comparative drop size distributions nor of the effect of air velocity on the space distribution (except for Nozzle *C*). Superficial examination of the sprays indicated that Nozzle *A* yielded the finest spray while Nozzles *D* and *E* yielded substantially coarser drops.

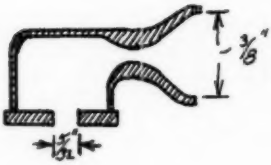
TOWER PERFORMANCE

In order to determine the actual performance of the tower with the various sprays installed, steady state runs were made on the counterflow side and the conductances computed. The results are shown in Fig. 6 in which the conductance for mass transfer is plotted against the liquid rate. From these curves the actual over-all effect of the various sprays is evident. The fact



1. Nozzle *A*

This nozzle has a machined circular inlet section, a machined mixing chamber, and a machined, conical orifice. The water is forced in on one side of the mixing chamber, thus giving the water a centrifugal motion. It is then forced downward and out through the orifice which is the same size as the inlet. The spray produced is extremely fine, but the spray cone angle is wide.

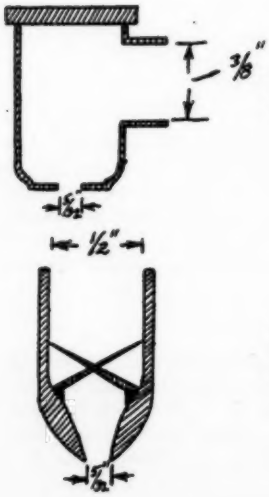


2. Nozzle *B*

This nozzle is the same as the above except that the conical orifice has been plugged with solder and a straight hole drilled through. The spray produced is not quite so fine and is not so evenly distributed about the spray cone. The spray cone angle, however, is reduced about ten degrees.

3. Nozzle *C*

A piece of copper tubing, flared at the end, is soldered to the orifice face of Nozzle A in order to cut down the spray angle. The desired result is obtained, the angle being reduced by about twenty degrees, but again the spray produced is much coarser than that produced by the original nozzle.



4. Nozzle *D*

Nozzle D has a large rough entrance section, a machined finish mixing chamber, and a thin cylindrical outlet orifice. The drops produced are rather large and unevenly distributed about the spray cone.

5. Nozzle *E*

This nozzle is designed to produce a flat spray which is evenly distributed across the cone. The drops produced are rather large, however. The nozzle consists of a straight entrance section, a movable piece forced to rotate by the liquid flow, and a mixing chamber.

FIG. 5. DESCRIPTION OF SPRAY NOZZLES

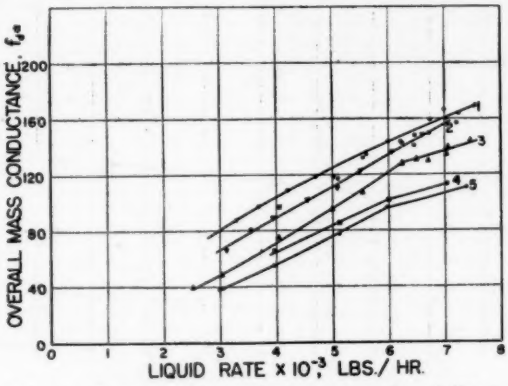


FIG. 6. OVER-ALL MASS CONDUCTANCE, f_{aa}, lb/ft^3 hr (lb/lb) VERSUS LIQUID RATE, lb/hr

Air rate = 5000 lb/hr. Legend:

1. Nozzle *A*
2. Nozzle *B* with cylindrical bore
3. Nozzle *C* with 1 in. nipple
4. Nozzle *D*
5. Nozzle *E*

that one of the lowest curves is that of Nozzle *D* in contrast to the point that it gave the best distribution is interesting for it indicates that the size of the drops is of greater importance than the space distribution of the water at outlet. The other curves of conductance versus water rate actually lie in the same relative order as the relative drop sizes, the implication being that the greatest portion of the cooling in the tower is due to the drops even though most of the

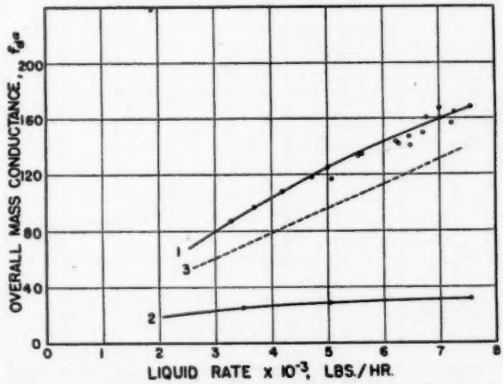


FIG. 7. OVER-ALL MASS CONDUCTANCE, f_{aa}, lb/ft^3 hr (lb/lb) VERSUS LIQUID RATE, lb/hr

Air rate equals approximately 5000 lb/hr. Legend:

1. Conductance for spray and walls combined
2. Conductance for walls alone
3. Conductance for sprays alone (obtained by subtraction)

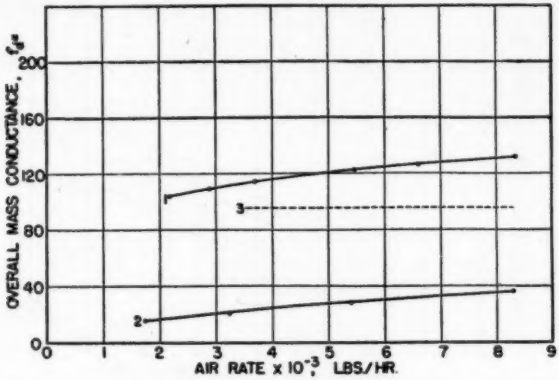


FIG. 8. OVER-ALL MASS CONDUCTANCE, f_{aa}, lb/ft³ hr (lb/lb) VERSUS AIR RATE, lb/hr

Liquid rate equals approximately 5000 lb/hr. Legend:
1. Conductance for spray and walls combined
2. Conductance for walls alone
3. Conductance for sprays alone (obtained by subtraction)

water passes down the walls. In other words, most of the cooling takes place at the section where the water first enters. This conclusion is further substantiated by recent results obtained on the parallel flow side of the tower which indicate that the over-all mass conductance for parallel flow is higher than that for counterflow (17). In the region of the spray nozzles, the air flow is probably appreciably distorted by the high velocity water jet and drops.

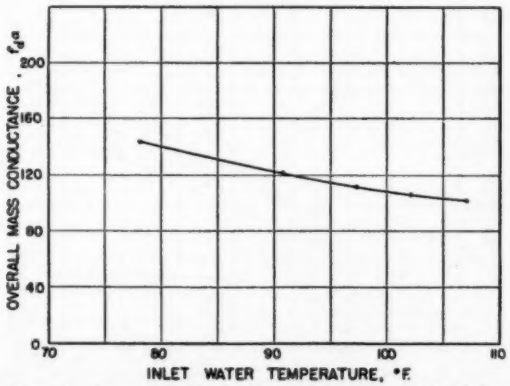


FIG. 9. OVER-ALL MASS CONDUCTANCE, f_{aa}, lb/ft³ hr (lb/lb) VERSUS INLET WATER TEMPERATURE, F

Liquid rate equals approximately 5000 lb/hr
Air rate equals approximately 5000 lb/hr

This distortion of the air flow accompanied by the high energy transfer rates, which occur in this region and which produce high temperature gradients in both fluids, probably vitiates to an appreciable degree many of the hypotheses incorporated in the rate coefficients as given by Equations (1), (2) and (3).

At higher magnitudes of the liquid rate, L, all the curves tend to break. In the case of the original Nozzle A a number of runs were made at high liquid rates to attempt to verify this change in slope. It was found that the points obtained scattered widely.

In order to determine how much cooling was done by the water falling down the walls of the tower the sprays were plugged and the water was distributed around the walls of the tower by means of drilled pipes in such manner that the sheet surface area exposed was approximately the same as that exposed when the sprays were operating. The liquid rate and air rate were varied and it was found that the mass conductance, $f_d a$, varied approximately as $G^{0.5}$ and $L^{0.3}$ (Figs. 7 and 8)

$$f_d a = 0.033 G^{0.5} L^{0.3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Sherwood and Holloway (15) obtained a somewhat similar equation for the absorption of ammonia in ring-packed towers

$$K_g a = 0.046 G^{0.5} L^{0.4} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

The exponent 0.5 for the air rate compares favorably with the exponent 0.48 obtained by London for mass transfer from streamlined packing sections (8). Gilliland and Sherwood report an exponent of 0.8 for wetted wall towers (3). However, in their work the flow of liquid film was viscous and consequently the relative velocity of air and water did not differ greatly from the absolute air velocity. In the present case the relative velocity was greatly different from the absolute. Relative velocities (approximate) were employed to correlate the spray tower data, and it was found that the conductance then varied approximately as the 0.8 power. It is impossible to make exact comparisons with results of other experimenters because of the nature of the construction of the tower, but it is believed that the results indicated are valid and that they are partially substantiated by the closeness of the comparisons given.

A complete series of runs was made using Nozzle A and varying the liquid and air rates. The results (Figs. 7 and 8) indicated that the mass conductance varied with $L^{0.76}$ and $G^{0.17}$.

$$f_d a = 0.19 L_0^{.76} \text{ (constant air rate and inlet water temperature)} \quad . \quad . \quad . \quad . \quad (10)$$

$$f_d a = 0.043 L^{0.76} G^{0.17} \text{ (constant inlet water temperature)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

If a thermal circuit is drawn it is seen that the conductances from the drops and from the sheets are in parallel and consequently are directly additive, subject to several conditions. First, for a given amount of water down the tower only a certain percentage passes down the walls; this connection may be made approximately by using the water distributions previously obtained. Second, and probably more important, when the spray is operating, the water from the walls is evaporating into air which contains water droplets and the potentials are therefore not so great as for sheet cooling alone. The greatest

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amount of material transfer takes place at the water entrance; consequently on the counterflow side it should be possible to assume that in the region below the level where the spray strikes the walls humidity conditions do not differ too widely from the case in which only sheet cooling occurs. Subtracting magnitudes on the sheet cooling curves (corrected for space water distribution) from magnitudes on the over-all cooling curves will then give the conductance from the spray alone. The results obtained after such procedure indicate that for the sprays the conductance varies linearly with the liquid rate and is relatively unaffected by changes in air rate. This result is con-

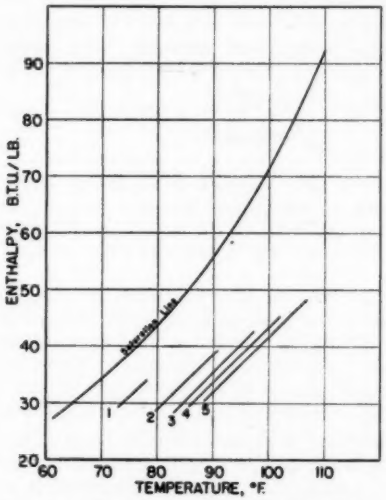

FIG. 10. EFFECT OF INLET WATER TEMPERATURE ON THE OPERATING LINE. ENTHALPY, BTU/LB VERSUS TEMPERATURE, F

Legend: 1. Run No. 66; 2. Run No. 38; 3. Run No. 63; 4. Run No. 64; 5. Run No. 65

firmed by the work of Hixson and Scott (5) who found that the coefficient for ammonia absorption in a spray tower with the wall effect removed varied directly with the liquid rate. The fact that the conductance from the drops is unaffected by changes in the air rate is readily explained in that the relative velocity of the drops and the air stream is dependent mainly on the water velocity (approximately 100 to 150 fps at 5,000 lb per hour) and only slightly on the air velocity (5 ft per second, maximum).

The temperature of the inlet water (for a given tower and air inlet conditions) has been found to have a profound effect upon the over-all mass transfer coefficient. Haslam, Hershey, and Keen (4) in their work on ammonia absorption found that increasing the temperature of the fluid caused the coefficient of absorption to decrease as the 1.4 power of the absolute temperature. Dwyer and Dodge (1) indicate that the coefficient of absorption varies in-

versely with temperature. In order to determine the effect in the cooling tower, several runs were made holding the liquid and air rates constant and changing only the inlet temperature of the water. The results are indicated in Fig. 9. When plotted on log-log paper this curve yields a slope of — 1.09. The equation for the conductance may now be written

$$f_a a = 5.89 \frac{L^{0.76} G^{0.17}}{T_1^{1.09}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

This equation holds only within the operating limits indicated in Figs. 6 to 9.

Whenever a liquid and gas come into contact there exists on the gas side of the interface a layer of gas in which motion by convection is slight compared to that in the main body of gas. Similarly on the liquid side there is a surface layer which is practically free from convection. These two layers offer resistance to the flow of heat and material. The resistances are dependent upon the thicknesses of the layers, which in turn are functions of the ratio of viscosity to density (7) (3). Since the viscosity and density are dependent upon the temperature, it was thought that the effect of temperature might be due to its effect on these variables. Kowalke, Hougen, and Watson (6) state that the transfer coefficient for ammonia decreases with increase in temperature in the same order of magnitude as the viscosity increase of the gas.

A search of the literature revealed that very little experimental work has been done on effect of moisture upon the viscosity of air. What little there is, however, (2) indicates that moisture does not change the viscosity appreciably. Gilchrist (2) obtained the following results for dry and saturated air:

Dry air (20.2 C) = 1.811×10^{-7} gms/cm. sec. (Average of four values which agreed within 0.3 per cent).

Sat. air (20.2 C) = 1.816×10^{-7} gms/cm. sec. (Average of three values which agreed within 0.3 per cent).

Calculation of the ratios of viscosities to densities, using viscosity of dry air and density of the air-water vapor mixture shows that the change with temperature does not account for the great change in over-all conductance with temperature.

Another possibility that presents itself is that the liquid side resistance might be appreciable. In the case of water running down the walls of the tower, the assumption of no resistance on the liquid side is probably justified, since the water is thoroughly mixed as it tumbles over the wall projections. The water drops, however, might present another picture. It has been suggested (10), (12) that as the drops fall through the tower they experience little internal mixing. Increased water temperature would augment the convection currents within the drops (if convection exists at all) as they fall and thus increase the internal mixing. The two effects, consequently, are opposed and probably offset each other. Finally the extent of small drop vaporization (8) will be changed by the water inlet temperature. In Fig. 10 the operating and equilibrium lines on a temperature-enthalpy diagram (8), (9), (13) are shown for the runs in which inlet water temperature alone was changed. It is readily seen that as the temperature is increased the change in enthalpy of the air passing through the tower is increased. It may also be noted that the operating line moves further away from the equilibrium line, *i.e.*, the driving potential becomes larger.

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In the calculation of the conductance, the conductance varies directly as the energy change of the air and inversely as the enthalpy potential.

$$f_h a = \frac{G}{V} \frac{\Delta h}{(h_1 - h)_{lm}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

If $(h_1 - h)_{lm}$ increases more rapidly than Δh as the temperature is increased, then the conductance will decrease. This, apparently, is what actually occurs.

LEWIS MODULUS

Various experimenters have found that the ratio of the conductance for heat

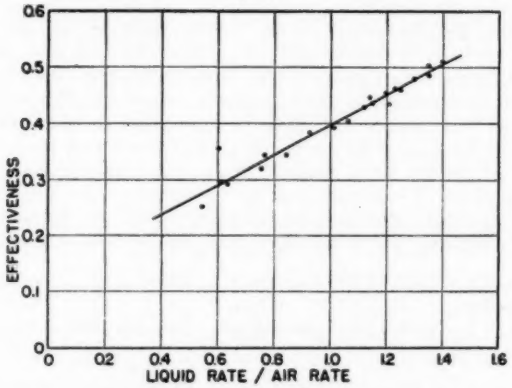

FIG. 11. EFFECTIVENESS, $\frac{h_1 - h_2}{h_{11} - h_2}$ VERSUS RATIO OF LIQUID RATE TO AIR RATE

Inlet water temperature = 90 to 100 F

transfer to the conductance for mass transfer is equal to the specific heat of the air stream for the air-water vapor system.

$$f_e/f_d = s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

If the transfer areas for mass and sensible heat are the same, then

$$f_e a/f_d a = s \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

It was found that for the spray tower tested this relationship does not often hold. The failure of the data to yield good checks of the Lewis Modulus (for runs in which the sprays were operating) is probably due to the complete vaporization of small drops. The consequence of this vaporization would be to increase the mass conductance, $f_d a$, and decrease the thermal conductance, $f_e a$, thus yielding low values of the Lewis Modulus, as found experimentally. It was also found that in the cases in which the Lewis Modulus did hold, the change in temperature of the air passing through the tower was comparatively

large. This merely means that the effect of small drop vaporization (on the thermal conductance) is negligible when the thermal conductance is large.

EFFECTIVENESS

The effectiveness of the tower, calculated on an enthalpy basis (8) and (19), is plotted as a function of the ratio of liquid rate to air rate in Fig. 11.

$$\epsilon_h = \frac{h_1 - h_2}{h_{11} - h_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

The equation of the average line drawn through the experimental points is

$$\epsilon = 0.125 + 0.27 \times \frac{L}{G} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

The plotted magnitudes have not been corrected for change in inlet water temperature which varied from 90 F to 100 F.

CONCLUSIONS

1. In a spray tower the performance is affected more by the drop size distribution than by the water distribution within the tower. The greatest amount of cooling is obtained in the section where the water first enters.

2. The conductances '$f_d a$' determined for this spray tower should not be extrapolated too widely to other tower volumes, for the drop size distribution and the liquid distribution in the tower are important variables which have not yet been included in the correlation.

3. The conductance for mass transfer from the walls of the tower varies as $G^{0.5}$ and $L^{0.3}$.

4. The over-all conductance for mass transfer from both spray and walls (*i.e.*, over the whole tower) may be represented by the empirical equation

$$f_d a = 5.89 \frac{L^{0.76}\, G^{0.17}}{T_1^{1.09}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

5. The mass conductance for the spray alone varies linearly with the liquid rate and is unaffected by the air rate.

6. Complete vaporization of small drops causes the ratio of $f_e a / f_d a$ to be smaller than the humid heat of the air stream.

ACKNOWLEDGMENT

The writers desire to express their appreciation to H. Eagles and S. Hori for their aid in the performance of the laboratory work, to the Golden Gate Chapter of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the San Francisco Air Conditioning Society for their interest in the work. The W. R. Ames Co., San Francisco, made the cooling tower available to the University of California.

NOMENCLATURE

a = Transfer area per unit volume of tower, ft²/ft³
c = unit heat capacity of liquid water, Btu/lb F

f_o = over-all unit thermal conductance for convection, Btu/ft² hr F
f_d = over-all unit conductance for diffusion, lb/ft² hr (lb/lb)
f_h = over-all unit conductance for enthalpy transfer, Btu/ft² hr (Btu/lb)
G = dry air rate, lb/hr
h = air-water vapor-mixture enthalpy, Btu/lb
Δh = change in enthalpy of air passing through tower, Btu/lb
H = air humidity, lb of water vapor/lb of bone dry air
ΔH = change in humidity of air passing through tower, lb/lb
L = liquid water flow rate, lb/hr
ΔL = change in basin level, ft
s = humid heat of air-water vapor mixture, Btu/F lb of bone dry air
t = air main body temperature, F
Δt = change in temperature of air passing through tower, F
t_{wb} = air main body wet bulb temperature, F
T = water main body temperature, F
V = tower volume = 35 ft³

NTU = number of transfer units = $\frac{f_d a V}{G}$

Subscripts

1 = tower conditions at water entrance and air exit
2 = tower conditions at water exit and air entrance
i = interface conditions
lm = logarithmic mean

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No. 1190

MERKEL'S COOLING DIAGRAM AS A PERFORMANCE CORRELATION FOR AIR-WATER EVAPORATIVE COOLING SYSTEMS

By H. B. NOTTAGE *

THERE can be no question as to the desirable and useful nature of a general performance correlation for air-water evaporative cooling systems which would involve the use of but a single process diagram or chart on which any operating line, as defined by a corresponding set of initial and terminal conditions and located according to the independent variables involved, would be a straight line that could be laid off in a direct and simple manner. A chart of this nature was devised several years ago by Friedrich Merkel (1)[1] in his fundamental study of evaporative cooling. It will be the aim of this paper to develop the principle of this method of analyzing and correlating performance data with the aid of Merkel's cooling diagram, and to apply the method developed to the study of a complete set of performance data obtained from tests on a forced-draft packed cooling tower.

BASIC EQUATIONS FOR AN EVAPORATIVE COOLING PROCESS

Consider an elementary cross-sectional volume within an adiabatic steady-state counterflow system, Fig. 1, wherein an exchange of heat and mass (energy) takes place spontaneously between a stream of water, at a temperature other than the adiabatic-saturation equilibrium temperature, and a stream of air and its associated water vapor for which the moisture content is below saturation (2, 3).

A mass balance on the elementary volume yields the relation:

$$dL = G dH \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

An energy balance for a steady-flow system in which the changes of potential and kinetic energy are negligible, and in which no external work or heat transfer is involved, becomes merely an enthalpy balance for the

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[1] Numerals in parentheses refer to Bibliography.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, San Francisco, Calif., June, 1941.

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interacting fluid streams. Restricting the analysis to conditions under which the air and vapor behave as perfect gases and the unit heat capacities may be taken as constant, the enthalpies involved, referred to a datum temperature T_0 and a saturated liquid at that temperature, may be expressed by:

$$\text{Dry air enthalpy} = c_a (t - T_0) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$\text{Water vapor enthalpy} = h_{fg_0} + c_s (t - T_0) \quad . \quad . \quad . \quad . \quad . \quad (3)$$

$$\text{Water liquid enthalpy} = c_L (T - T_0) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Carrying through the energy balance on the elementary exchanger volume, there results the equation:

$$Lc_L dT + (dL)c_L (T + dT - T_0) = h_{fg_0} GdH + G(c_a + Hc_s)\ dt + Gc_s(dH)(t + dt - T_0) \quad . \quad (5)$$

After simplifying the Equation (5), neglecting differentials of the second

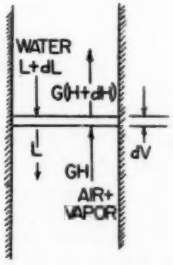

FIG. 1. THE EVAPORATIVE COOLING SYSTEM

order, and substituting the unit heat capacity at constant pressure of the air-vapor mixture, s, for the factor $(c_a + Hc_s)$, there is obtained:

$$Lc_L dT + (dL)c_L(T - T_0) = h_{fg_0} GdH + Gsdt + Gc_s(dH)(t - T_0) \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The two terms on the left side of Equation (6) represent the rate of change of enthalpy for the water stream in passing through the elementary exchanger volume, and the three terms on the right represent the corresponding rate of change of enthalpy for the air-vapor stream.

Transfer-rate equations are necessary to complete the analysis of the process. The rate of convective heat transfer from the main water body to the air-vapor stream in the elementary volume considered may be expressed as:

$$dq_c = f_c(T - t)adV \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Radiant heat transfer is negligible in an enclosed exchange system if adjacent surfaces are essentially at the same temperature.

The rate of convective mass transfer between the two streams may be taken as proportional to the difference in specific humidity between the saturated air-vapor mixture at the temperature of the main water body and

the actual gas-stream mixture within the elementary exchanger volume, $(H_L - H)$. It then follows that the rate of heat removal from the main water body due to the evaporation of a portion of the liquid stream, which must be equal to the rate of change in enthalpy for the evaporating stream between the main body of liquid and the liquid-gas interface (beyond which point changes in the state of the evaporating liquid stream have only an indirect bearing upon the energy removed from the liquid insofar as the interface conditions are influenced thereby), may be expressed in the form:

$$dq_D = f_D(H_L - H)(adV)[h_{tg_T} + c_s(T - T_1)], \quad . \quad . \quad . \quad . \quad . \quad (8)$$

where T_1 is the interface temperature.

Since the process envisioned is adiabatic, it must follow that:

$$dq_e + dq_D = Lc_L dT + (dL)c_L(T - T_o) = h_{tg_o} GdH + Gsdt + Gc_s(dH)(t - T_o). \quad . \quad (9)$$

It will next be advisable to anticipate that the relationship ultimately desired is one in terms of enthalpies, primarily because in such terms the analysis reduces to a convenient and simple form. The enthalpy of the air-vapor mixture at temperature T and of moisture content H_L, which is the saturated mixture in equilibrium with the main water body, may be expressed as:

$$h_L = c_s T + H_L[h_{tg_o} + c_s(T - T_o)] = s_L T + H_L(h_{tg_o} - c_s T_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

In a similar manner, the enthalpy of the mixture at the conditions prevailing in the main body of the gas stream is:

$$h = st + H(h_{tg_o} - c_s T_o) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10a)$$

Further, let it be assumed that the over-all unit surface conductances for convective heat transfer and for convective mass transfer may be inter-related to a sufficient degree of accuracy by the Lewis equation (2, 5),

$$f_e \cong s f_D, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

and that to a very good approximation $s_L \cong s$.

On this basis, the total rate of energy transfer between the interacting streams becomes, after combining Equations (7) and (8):

$$dq_e + dq_D = f_D(adV)\left[s(T - t) + (H_L - H)[h_{tg_T} + c_s(T - T_1)]\right] \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Substituting for the latent heat in terms of Kirchoff's equation (4),

$$h_{tg_T} = h_{tg_o} + (c_s - c_L)(T - T_o),$$

collecting terms, and introducing Equations (10) and (10a),

$$dq_e + dq_D = f_D(adV)\left[(h_L - h) + (H_L - H)[c_s(2T - T_1) - c_L(T - T_o)]\right] \quad . \quad . \quad (13)$$

Then, taking the first equality of Equation (9) and there replacing dL by its equivalent, $f_D(adV)(H_L - H)$, Equation (13) becomes:

$$Lc_L dT = f_D(adV)\left[(h_L - h) + (H_L - H)[c_s(2T - T_1) - 2(c_L T - T_o)]\right] \quad . \quad . \quad . \quad (14)$$

The numerical magnitude of the factor involving $(H_L - H)$ in Equation

(14) will usually be relatively small, so that it will be permissible, especially in view of the resulting advantages of the simplified form of the equation, to merely drop that factor from the equation. The resulting slight error, as well as the error due to the use of the Lewis equation and the assumption that $s_L \cong s$, will be thrown into the magnitude of the conductance f_D to be experimentally determined; and this conductance, in turn, will then no longer be called f_D, but will be redefined as a conductance based upon an enthalpy potential, f_h. The form of Equation (14) to be used in later applications thus becomes:

$$L_{el}dT = f_h(adV)(h_L - h) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Another reasonable simplification, which leads to a quite useful approximate relationship where the error will ordinarily not exceed two or three per cent, is to neglect the rate of water evaporation in comparison to the total rate of water flow through the system. Equation (6) then may be written as:

$$Lc_L dT \cong Gdh \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16)$$

Application of the preceding equations may be carried out in any one of several directions according to the problem under consideration. The further development to follow will deal with Equations (15) and (16), for they form the basis of Merkel's cooling diagram.

PRINCIPLE OF MERKEL'S COOLING DIAGRAM

Admitting that a suitable over-all mean value for the enthalpy potential in Equation (15) may be defined, and treating f_h as a constant, integration of the differential equation yields:

$$Lc_L(T_1 - T_2) = f_h aV(h_{L_m} - h_m) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

And from Equation (16), it follows after integration that, to a close approximation:

$$Lc_L(T_1 - T_2) = G(h_1 - h_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

The mean potential in Equation (17) may properly be expressed as a difference of two mean values, for:

$$(h_L - h)_m = \frac{1}{aV}\int_0^{aV} (h_L adV - hadV) = h_{L_m} - h_m$$

Since h_L is taken as the saturation enthalpy at the temperature of the main water body, which enthalpy is determined only by that temperature for pressures near atmospheric, and since the water temperature varies continuously with the transfer area swept over, the mean value of h_L with respect to transfer area will be closely equivalent to the mean with respect to water temperature, *viz.*:

$$h_{L_m} = \frac{1}{(T_1 - T_2)}\int_{T_2}^{T_1} h_L dT \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

The inaccuracy involved in this substitution is of the same order as that accompanying the use of the logarithmic-mean enthalpy potential (9, 10).

A directly parallel simplification is not permissible for h_m, since it is a function of both t and H, which vary interdependently with the water temperature in a rather complicated manner (7), thereby making strict evaluation of the integral an extremely tedious process. However, to a good first approximation, the variation of the gas-stream enthalpy over the low-temperature range ordinarily encountered may be taken as linear with both quantities, so that:

$$h_m \cong \frac{h_1 + h_2}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

The effect of this simplification will later be investigated in detail in connection with the illustrative example of cooling tower performance analysis

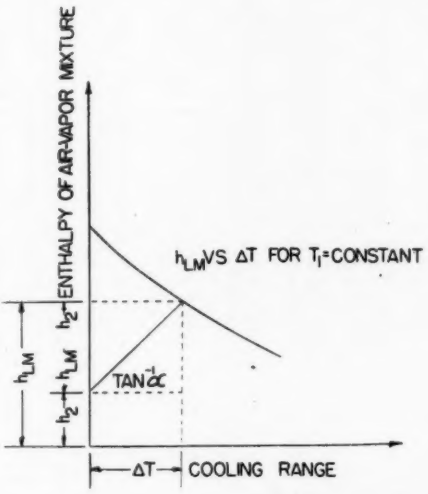

FIG. 2. PRINCIPLE OF THE COOLING DIAGRAM

to be undertaken. Justification for the step must be established and accepted upon an empirical basis, for it is necessary to use the arithmetic mean in order to bring about the whole simplified form of Merkel's method of analysis.

From Equation (18):

$$h_1 = h_2 + \frac{\Delta T}{\lambda},$$

denoting the ratio G/L by λ and the cooling range $(T_1 - T_2)$ by ΔT.

Then, from Equations (20) and (17),

$$\rho_1 a V(h_{t_m} - h_2 - \frac{\Delta T}{2\lambda}) = Lc_1\Delta T,$$

TABLE I—MEAN ENTHALPIES OF SATURATED AIR FOR THE

	MEAN ENTHALPY, h_{L_m}										
T_2 F	T_1 = 170 F	T_1 = 165 F	T_1 = 160 F	T_1 = 155 F	T_1 = 150 F	T_1 = 145 F	T_1 = 140 F	T_1 = 135 F	T_1 = 130 F	T_1 = 125 F	T_1 = 120 F
170	530.6										
165	485.7	444.1									
160	447.2	408.6	375.6								
155	413.7	377.8	346.9	320.1							
150	384.5	350.8	321.8	296.7	274.7						
145	358.6	326.9	299.6	276.1	255.2	236.8					
140	335.6	305.6	279.9	257.6	237.9	220.6	205.2				
135	315.0	286.6	262.2	241.0	222.4	206.0	191.5	178.4			
130	296.5	269.5	246.3	226.1	208.5	192.9	179.1	166.7	155.7		
125	279.7	254.0	231.9	212.7	195.9	181.1	167.9	156.1	145.4	136.1	
120	264.5	239.9	218.8	200.5	184.5	170.3	157.8	146.5	136.4	127.5	119.3
115	250.6	227.1	206.9	189.4	174.1	160.4	148.6	137.9	128.2	119.7	111.8
110	237.9	215.4	196.1	179.3	164.6	151.7	140.2	130.0	120.8	112.5	105.1
105	226.3	204.7	186.1	170.0	156.0	143.5	132.5	122.7	114.0	106.0	98.9
100	215.6	194.9	177.0	161.5	148.0	136.1	125.5	116.1	107.6	100.1	93.2
95	205.7	185.7	168.5	153.7	140.7	129.2	119.0	110.0	101.9	94.5	88.0
90	196.5	177.3	160.8	146.4	133.9	122.9	113.1	104.4	96.6	89.6	83.3
85	188.1	169.5	153.6	139.7	127.7	117.0	107.6	99.2	91.7	85.0	78.9
80	180.2	162.2	146.8	133.5	121.8	111.6	102.5	94.4	87.2	80.7	74.9
75	172.9	155.5	140.6	127.7	116.4	106.5	97.8	90.0	83.0	76.7	71.1
70	166.0	149.2	134.8	122.3	111.4	101.8	93.4	85.8	79.1	73.0	67.6
65	159.7	143.4	129.4	117.3	106.8	97.5	89.3	82.0	75.5	69.6	64.4
60	153.7	137.9	124.3	112.6	102.4	93.4	85.5	78.4	72.1	66.4	61.4
55	148.1	132.7	119.6	108.2	98.3	89.6	81.9	75.0	68.9	63.5	58.5
50	142.8	127.9	115.1	104.1	94.5	86.0	78.4	71.9	66.0	60.7	55.9
45	137.9	123.4	111.0	100.2	90.9	82.7	75.2	69.0	63.2	58.1	53.5
40	133.2	119.1	107.0	96.6	87.5	79.5	72.5	66.2	60.6	55.6	51.1
35	128.8	115.1	103.3	93.2	84.3	76 5	69.7	63.6	58.2	53.3	48.9

or,

$$h_{L_m} - h_2 = c_L \Delta T \left(\frac{1}{2\lambda} + \frac{L}{f_h a V} \right) \quad . \quad . \quad . \quad . \quad . \quad (21)$$

Defining the *cooling factor*, α, as

$$\alpha = \frac{h_{L_m} - h_2}{c_L \Delta T} = \left(\frac{1}{2\lambda} + \frac{L}{f_h a V} \right) \quad . \quad . \quad . \quad . \quad . \quad (22)$$

a very useful graphical representation based upon this definition may be developed. If, as in Fig. 2, h_{L_m} is plotted as a function of ΔT for a given constant value of T_1 and a straight line is drawn from the point (O, h_2) to the point $(\Delta T, h_{L_m})$ on the curve for the value of T_1 prevailing, the line will have a slope of magnitude α (remembering that c_L is unity for water). The minimum value that this slope can have at any time is:

$$\alpha_{min} = \frac{1}{2\lambda} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

corresponding to $f_h = \infty$ for finite air and water rates. Therefore, a possible measure of the effectiveness of a cooling process is the ratio:

TEMPERATURE RANGE T_1 TO T_2. BAROMETRIC PRESSURE = 29.92 IN. HG

BTU/LB DRY AIR

T_1 = 115 F	T_1 = 110 F	T_1 = 105 F	T_1 = 100 F	T_1 = 95 F	T_1 = 90 F	T_1 = 85 F	T_1 = 80 F	T_1 = 75 F	T_1 = 70 F	T_1 = 65 F	T_1 = 60 F	T_1 = 55 F	T_1 = 50 F
104.8													
98.3	92.1												
92.4	86.5	81.2											
87.0	81.4	76.3	71.6										
82.1	76.7	71.8	67.3	63.2									
77.6	72.4	67.7	63.4	59.4	55.8								
73.4	68.4	63.9	59.8	56.0	52.5	49.3							
69.6	64.7	60.4	56.5	52.9	49.6	46.4	43.6						
66.0	61.4	57.2	53.4	49.9	46.7	43.7	41.0	38.5					
62.7	58.2	54.2	50.5	47.2	44.0	41.2	38.6	36.2	34.0				
59.6	55.3	51.4	47.9	44.7	41.6	38.9	36.4	34.1	32.0	30.0			
56.8	52.6	48.8	45.4	42.3	39.4	36.8	34.4	32.1	30.1	28.2	26.4		
54.1	50.1	46.4	43.1	40.1	37.3	34.8	32.5	30.3	28.3	26.5	24.8	23.2	
51.6	47.7	44.2	41.0	38.1	35.4	32.9	30.7	28.6	26.7	24.9	23.3	21.7	20.3
49.3	45.5	42.1	39.0	36.1	33.6	31.2	29.0	27.0	25.1	23.4	21.8	20.3	18.9
47.1	43.4	40.1	37.1	34.3	31.8	29.6	27.4	25.5	23.7	22.0	20.5	19.0	17.7
45.0	41.5	38.2	35.3	32.7	30.2	28.0	25.9	24.0	22.3	20.7	19.2	17.8	16.5

$$\frac{\alpha_{min}}{\alpha} = \frac{1}{2\lambda\alpha}.$$

This ratio will henceforth be called *Merkel's effectiveness.*

Generalizing the graphical construction indicated by the example presented, if a family of curves of h_{ι_m} *vs.* ΔT were to be plotted for a series of successive values of T_1, the resulting figure would be a *cooling diagram* which would serve as a rapid solution of Equation (22) relating L, G, f_h, ΔT, h_2, and α. A cross-plot of h_{ι_m} *vs.* ΔT along lines of constant T_2 would further serve to increase the usefulness of the diagram, and a suitable scale for the cooling factor, α, to be used in locating or determining the slope of *operating lines,* would also expedite the solution of problems on the diagram.

The data needed to plot such a cooling diagram are summarized in Table 1. The diagram itself is presented as Fig. 3 for the low temperature range only. Enthalpies on which the diagram was based were taken from the recent tabulation by Goodman (8). Perhaps it would be well to emphasize that the diagram is strictly correct for standard atmospheric pressure only, although no serious error will be involved for pressures within a reasonable range from the standard.

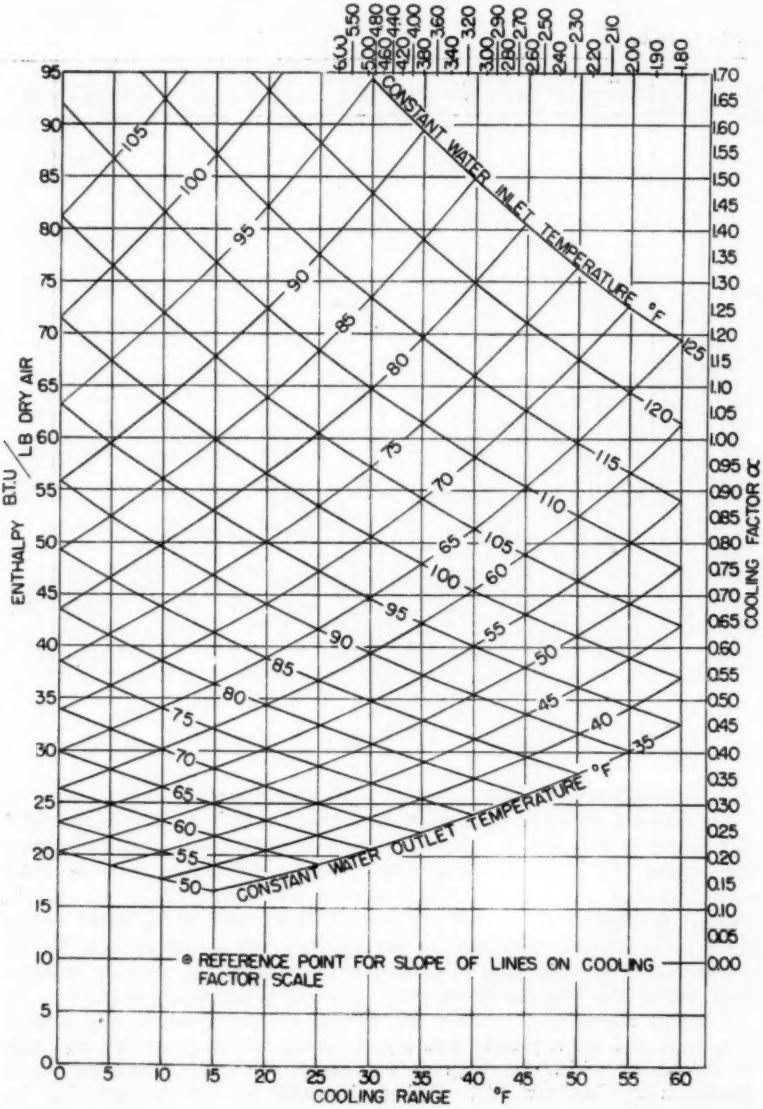
ENTHALPY B.T.U. / LB DRY AIR
COOLING FACTOR α
CONSTANT WATER INLET TEMPERATURE °F
CONSTANT WATER OUTLET TEMPERATURE °F
COOLING RANGE °F
⊕ REFERENCE POINT FOR SLOPE OF LINES ON COOLING FACTOR SCALE

FIG. 3. COOLING DIAGRAM FOR THE LOW TEMPERATURE RANGE

APPLICATION OF COOLING DIAGRAM TO ANALYSIS OF DATA ON A FORCED-DRAFT COOLING TOWER

By way of an example in the use of the cooling diagram as a primary means of studying the behavior of a cooling tower, the data of London, Mason, and Boelter (5) on a forced-draft packed tower will be analyzed to determine magnitudes of the cooling factor, α, and the overall average energy conductance, f_h. Possible methods of correlating these basic quantities will be considered and put to test, with a view to determining means of interpolating between measured conditions so as to facilitate performance prediction over the range of the available data.

In Merkel's study, he found that, for forced-draft towers where the packing, if used, was wetted in a uniform and constant manner, a good correlation could be obtained by plotting the cooling factor as a function of the air rate for each constant water rate. Any effect of the initial state of each fluid stream upon the cooling factor would presumably enter into the magnitude of the conductance f_h, although such an effect would not be expected to be noticeable except over a quite wide range of operating conditions. Merkel also demonstrated that it was possible to use the method of the cooling diagram in extrapolating from one system to another similar one to a limited extent. In the case of natural-convection towers, where the air rate is determined by the combined effect of the wind velocity and the natural buoyant forces, Merkel found that the cooling factor followed an approximate linear variation with the enthalpy of the inlet air for each constant wind velocity.

It is to be recognized that, for low water rates in packed towers, an incomplete wetting of the packing surface takes place. When this occurs, it is necessary to regard the effective transfer area as the fraction of the total which is wetted, provided only that the packing does not have a high thermal conductivity or a high porosity, either one of which would distort the balance between heat and mass transfer.

The data analyzed and the results obtained are summarized in Table 2, with the run numbers as given in the original paper. Magnitudes of the cooling factor were obtained graphically from a cooling diagram plotted to a still larger scale than the one in Fig. 3. Values of the mass-transfer conductance based upon the logarithmic-mean specific humidity potential, f_D, and the fraction of the total packing surface area wetted, η, were taken directly from the original paper. The total packing surface area for the tower in question, aV, was 418 square feet.

An indication of the effect of the slightly inaccurate nature of the Lewis equation and the simplifying approximations in the analysis is to be found in the extent to which the ratio f_h/f in Table 2 departs from unity. Merkel obtained an average value of 1.11 for this ratio as applying to his experimental tower packed with Raschig rings and having a spray-nozzle distribution arrangement, while the average of the tabulated values for the tower here considered is 0.95. It would appear that for preliminary design purposes where inadequate data are available, a value of unity could be assumed for this ratio.

The correlation of the cooling factor, α, as a function of the air and water rates is presented in Fig. 4. It is seen that the results of the analysis are quite satisfactory and serve to definitely establish the validity of the method

TABLE 2—ANALYSIS OF PERFORMANCE DATA ON A FORCED-DRAFT, PACKED, COUNTERFLOW COOLING TOWER

RUN NO.	t_2 F	H_2 milli.-lb / lb	h_2 Btu / lb dry air	T_1 F	T_2 F	ΔT F	α	G lb/hr	L lb/hr	$\lambda = G/L$	L/aV lb/hr ft²	f_h Btu / hr ft² Btu/lb	α min / α	f_D lb / hr ft² lb/lb	f_h/f_D	η FRACTION WETTED	f_h/η Btu / hr ft² Btu/lb
1	65.2	8.7	25.15	89.8	75.8	14.0	1.56	2970	3600	0.824	8.61	9.06	0.389	9.3	0.98	0.71	12.75
2	65.7	8.7	25.28	88.4	74.1	14.3	1.52	3460	3600	0.961	8.61	8.61	0.335	9.9	0.87	0.71	12.1
3	66.5	9.0	25.78	86.6	72.4	14.2	1.24	4250	3600	1.180	8.61	10.5	0.342	11.0	0.96	0.71	14.8
4	64.5	8.8	25.08	82.5	67.8	14.7	0.945	6590	3600	1.831	8.61	12.8	0.289	14.7	0.87	0.71	18.05
5	66.2	9.3	26.02	85.1	67.8	17.3	0.821	8430	3600	2.342	8.61	14.2	0.260	15.0	0.95	0.71	20.0
6	66.3	9.5	26.26	86.3	67.6	18.7	0.780	10120	3600	2.81	8.61	14.3	0.228	16.5	0.87	0.71	20.15
8	67.8	12.2	29.59	90.7	76.9	13.8	1.365	4850	4500	1.078	10.77	11.95	0.340	10.8	1.11	0.79	16.85
9	67.4	12.3	29.58	86.2	74.3	11.9	1.22	5260	4500	1.168	10.77	13.6	0.351	13.5	1.01	0.79	19.15
10	68.5	12.0	29.53	89.4	73.5	15.9	1.01	6010	4500	1.336	10.77	16.8	0.370	15.9	1.06	0.79	23.7
11	72.2	12.4	30.90	90.6	74.9	15.7	1.045	6670	4490	1.486	10.74	15.2	0.323	15.0	1.01	0.79	21.4
12	68.0	10.9	28.22	83.7	70.5	13.2	0.954	7400	4500	1.644	10.77	16.55	0.319	18.6	0.89	0.79	23.25
13	70.6	12.1	30.18	90.2	80.3	9.9	1.99	3240	5430	0.597	13.00	11.3	0.420	12.3	0.92	0.87	15.9
14	70.9	12.0	30.13	88.2	78.2	10.0	1.73	4070	5420	0.751	12.98	12.25	0.385	13.0	0.94	0.87	17.25
15	73.8	9.7	28.33	89.0	76.1	12.9	1.425	5150	5430	0.949	13.00	14.45	0.370	16.4	0.88	0.87	20.4
17	73.0	13.2	31.98	86.8	75.4	11.4	1.15	6480	5440	1.191	13.02	17.85	0.365	17.1	1.04	0.87	25.2
18	70.3	9.8	27.57	85.2	71.5	13.7	1.05	6500	5410	1.201	12.95	20.5	0.396	19.4	1.06	0.87	28.9
19	69.6	11.7	29.49	84.9	72.9	12.0	1.08	7270	5430	1.340	13.00	18.3	0.345	20.2	0.91	0.87	25.8
20	67.6	12.8	30.21	84.4	73.0	11.4	1.07	7330	5440	1.348	13.02	18.6	0.347	18.7	0.99	0.87	26.2
22	64.4	7.7	23.84	87.2	74.6	12.6	1.675	4590	6250	0.735	14.95	15.0	0.406	16.5	0.91	1.00	15.0
23	69.7	13.2	31.15	88.4	78.2	10.2	1.60	5360	6300	0.851	15.08	14.9	0.367	16.0	0.93	...	...
25	68.3	9.5	26.76	88.4	73.5	14.9	1.22	6540	6310	1.035	15.11	20.4	0.395	19.7	1.04	...	...
26	66.1	10.7	27.54	87.3	73.2	14.1	1.185	7020	6270	1.120	15.00	20.3	0.376	17.6	1.15	...	...
28	69.8	9.8	27.47	103.4	88.6	14.8	2.56	2880	7200	0.400	17.22	13.15	0.488	12.2	1.08	...	...
29	73.4	8.9	27.33	87.9	79.6	8.3	1.975	3420	7200	0.475	17.22	18.65	0.533	14.1	1.32	...	...
30	72.7	10.5	28.93	92.3	81.4	10.9	2.11	4280	7240	0.591	17.31	13.75	0.401	14.7	0.94	...	...
31	73.8	9.8	28.43	90.9	78.9	12.0	1.75	5400	7200	0.750	17.22	15.95	0.381	17.1	0.93	...	...
32	73.9	8.7	27.25	89.3	76.4	12.9	1.525	6340	7240	0.876	17.31	18.15	0.375	17.3	1.05	...	...

TABLE 2—ANALYSIS OF PERFORMANCE DATA ON A FORCED-DRAFT, PACKED, COUNTERFLOW COOLING TOWER
(*Concluded*)

33	67.4	9.2	26.21	82.5	71.8	10.7	1.365	8420	7240	1.163	17.31	18.5	0.315	20.6	0.90	...	...
34	68.6	9.4	26.72	82.0	70.9	11.1	1.215	10050	7240	1.390	17.31	20.2	0.296	22.0	0.92	...	...
35	67.5	10.0	27.11	103.4	91.8	11.6	3.51	2880	9180	0.314	21.95	11.4	0.453	12.2	0.93	...	...
36	72.3	9.2	27.41	87.8	80.8	7.0	3.11	3380	9220	0.367	22.05	12.6	0.438	14.2	0.89	...	...
37	70.3	9.9	27.69	93.2	83.4	9.8	2.66	4210	9070	0.464	21.7	13.7	0.405	14.4	0.95	...	...
38	71.1	8.8	27.67	92.1	80.8	11.3	2.19	5360	9250	0.580	22.1	16.6	0.393	16.3	1.02	...	...
39	66.3	9.4	26.16	81.6	74.1	7.5	2.04	6230	9220	0.675	22.05	16.95	0.363	18.0	0.94	...	...
40	70.7	8.6	26.28	88.4	78.3	10.1	2.11	6480	9250	0.701	22.1	15.8	0.348	20.9	0.76	...	...
41	67.6	9.6	26.71	84.8	75.3	9.5	1.80	8350	9220	0.905	22.05	17.65	0.307	20.7	0.85	...	...
42	69.1	9.1	26.54	84.5	74.2	10.3	1.60	10050	9220	1.090	22.05	19.35	0.287	21.6	0.90	...	...
43	72.0	9.1	27.23	90.3	84.5	5.8	4.36	2950	10840	0.272	25.95	10.3	0.421	12.5	0.82	...	...
44	65.8	9.2	25.81	94.8	86.9	7.9	3.97	3020	10660	0.283	25.5	11.6	0.445	12.5	0.93	...	...
45	71.5	9.2	27.22	90.4	83.8	6.6	3.77	3460	10840	0.319	25.95	11.8	0.415	13.4	0.88	...	...
46	69.1	8.8	26.21	88.8	81.6	7.2	3.27	4180	10840	0.385	25.95	13.15	0.397	15.1	0.87	...	...
47	75.1	8.6	26.58	89.8	81.2	8.6	2.72	5440	10800	0.504	25.85	14.95	0.365	16.4	0.91	...	...
48	64.8	8.1	24.38	86.8	78.6	8.2	2.72	5790	11160	0.519	26.7	15.15	0.355	16.7	0.91	...	...
49	68.2	9.3	26.51	90.3	81.2	9.1	2.63	6080	10760	0.565	25.7	14.5	0.337	17.7	0.82	...	...
50	75.6	9.2	28.23	90.3	80.8	9.5	2.31	6440	10800	0.596	25.85	17.55	0.363	18.7	0.94	...	...
51	63.6	8.2	24.21	85.6	76.9	8.7	2.40	6480	11160	0.581	26.7	17.35	0.358	18.3	0.95	...	...
52	67.7	9.7	26.85	85.5	77.0	8.5	2.13	8350	10760	0.776	25.7	17.35	0.303	20.4	0.85	...	...
53	67.1	9.0	26.03	82.1	74.1	8.0	1.965	10120	10840	0.934	25.95	18.05	0.272	22.8	0.79	...	...
54	66.3	9.4	26.18	94.5	88.1	6.4	4.94	2950	12490	0.236	29.9	10.6	0.430	11.6	0.91	...	...
55	72.6	9.4	27.70	88.4	83.4	5.0	4.58	3420	12530	0.273	30.0	10.9	0.400	12.7	0.86	...	...
56	67.2	9.3	26.27	90.0	83.7	6.3	4.03	4320	12560	0.345	30.05	11.65	0.359	13.8	0.84	...	...
57	66.9	9.0	25.87	89.6	82.5	7.1	3.49	5400	12600	0.428	30.1	12.95	0.335	15.1	0.86	...	...
58	65.8	9.1	25.72	93.0	83.7	9.3	3.03	6440	12530	0.513	30.0	14.65	0.322	15.6	0.94	...	...
59	69.2	8.9	26.33	86.9	79.0	7.9	2.62	8310	12460	0.667	29.8	15.95	0.286	19.3	0.83	...	...
60	69.9	9.0	26.61	84.1	76.6	7.5	2.33	9970	12530	0.795	30.0	17.65	0.270	22.0	0.80	...	...

despite the simplifying approximations. As far as the over-all performance of a given system operating within the range of available test data would be concerned, it would not really be necessary to pursue the analysis beyond obtaining magnitudes of the cooling factor. But, for comparison with other basic data and for extrapolation to other similar systems, knowledge of the over-all average conductance for energy transfer, f_h, would be needed as the

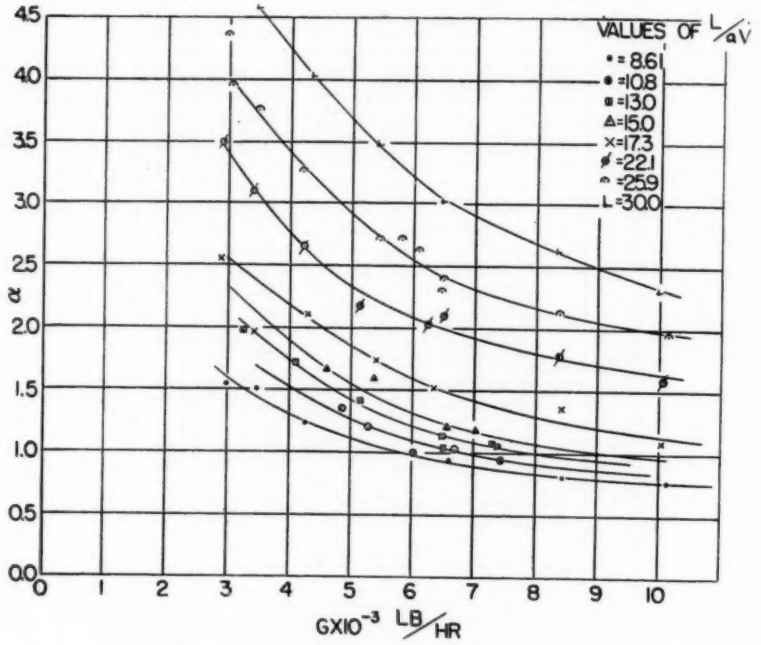

FIG. 4. VARIATION OF THE COOLING FACTOR WITH THE AIR RATE FOR CONSTANT WATER RATES

Key

L/aV = 8.61 lb/hr ft³	L/aV = 17.3 lb/hr ft³
L/aV = 10.8 lb/hr ft³	L/aV = 22.1 lb/hr ft³
L/aV = 13.0 lb/hr ft³	L/aV = 25.9 lb/hr ft³
L/aV = 15.0 lb/hr ft³	L/aV = 30.0 lb/hr ft³

most satisfactory basis of procedure. Were this conductance as computed by Merkel's method truly independent of the water rate, the cooling factor at constant air rate and with complete wetting of the packing would be directly proportional to the water rate, and the data should yield a single curve of the product $\alpha\lambda$ as a function of the air rate, G. Fig. 5 shows such a plotting.

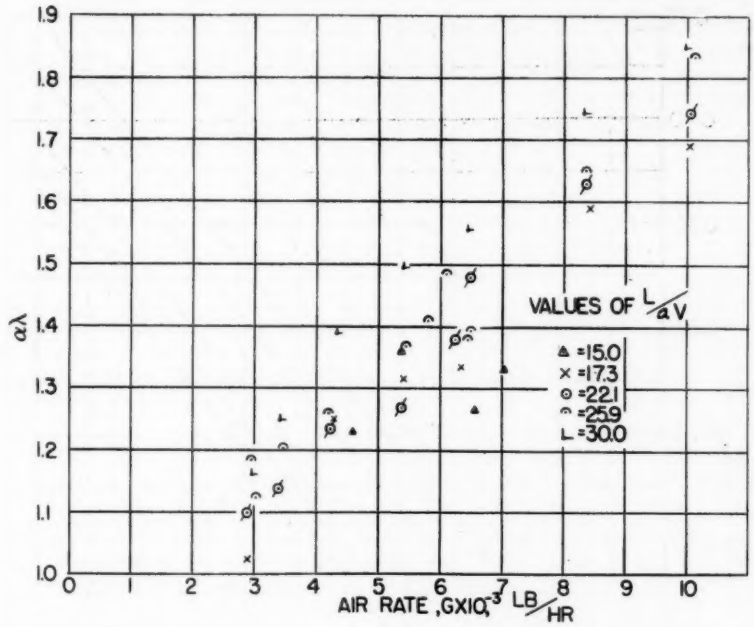

FIG. 5. VARIATION OF THE PRODUCT $\alpha\ \lambda$ WITH THE AIR RATE AT WATER RATES SUFFICIENT TO GIVE COMPLETE WETTING OF THE PACKING SURFACE

Key

L/aV = 8.61 lb/hr ft³	L/aV = 17.3 lb/hr ft³
L/aV = 10.8 lb/hr ft³	L/aV = 22.1 lb/hr ft³
L/aV = 13.0 lb/hr ft³	L/aV = 25.9 lb/hr ft³
L/aV = 15.0 lb/hr ft³	L/aV = 30.0 lb/hr ft³

where a small but noticeable trend with the water rate is evident; therefore, the water rate must have an influence upon the conductance f_h. It will be necessary to investigate this matter further in order to establish a proper interpolation equation for the effectiveness of the tower in question.

The form in which the effectiveness ratio may be expressed follows as:

$$\frac{\alpha_{min}}{\alpha} = \frac{\frac{L}{2G}}{\left(\frac{L}{2G} + \frac{L}{f_h \eta a V}\right)} = \frac{1}{1 + \frac{2G}{f_h \eta a V}} = \frac{1}{1 + \frac{2}{(NTU)\eta}} \quad . \quad . \quad . \quad (24)$$

where the introduction of the fraction of the total packing surface wetted, η, is necessary to provide a common basis for the comparison of all data. The

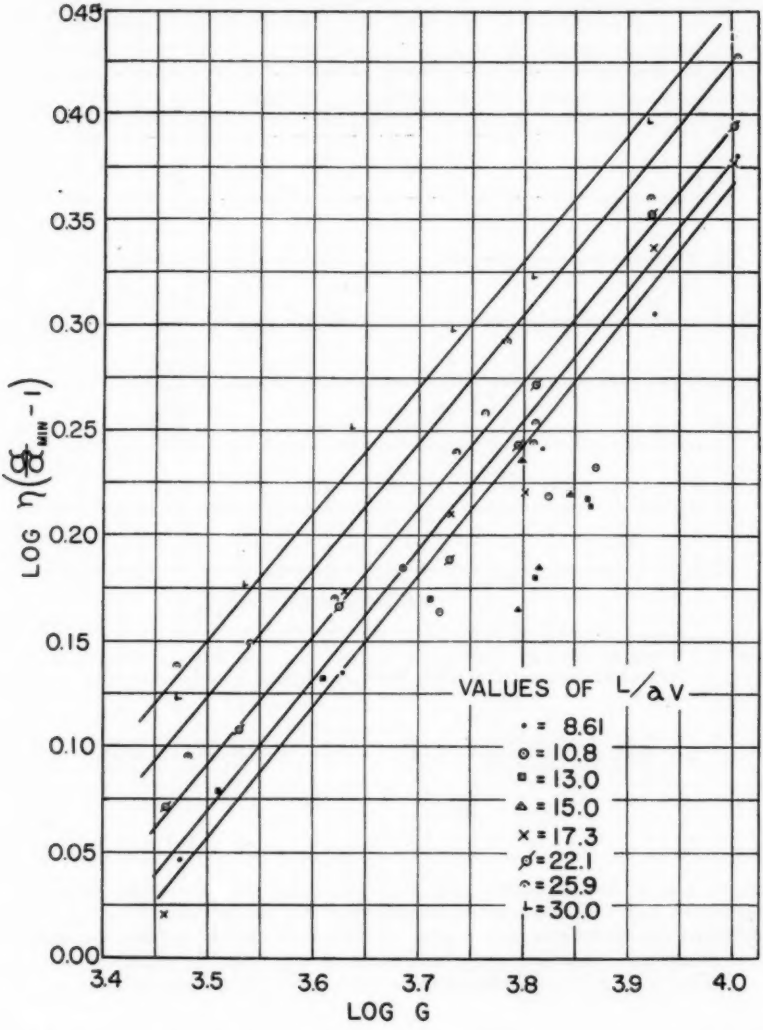

FIG. 6. LOGARITHMIC PLOT OF $\eta\left(\frac{\alpha-1}{\alpha_{min}}\right)$ *vs.* G AS A DEMONSTRATION OF THE EFFECT OF THE WATER RATE UPON THE OVERALL AVERAGE ENERGY CONDUCTANCE

Key

L/aV = 8.61 lb/hr ft²
L/aV = 10.8 lb/hr ft²
L/aV = 13.0 lb/hr ft²
L/aV = 15.0 lb/hr ft²
L/aV = 17.3 lb/hr ft²
L/aV = 22.1 lb/hr ft²
L/aV = 25.9 lb/hr ft²
L/aV = 30.0 lb/hr ft²

variation of η with the water rate (5) has been established as indicated in Table 2.

With reference to the conclusion drawn from Fig. 5, it becomes necessary to regard the over-all resistance to energy transfer from the main liquid body to the main gas body, $1/f_h$, as being composed of two series resistances; one representing the resistance to energy transfer from the main liquid body to the interface, and the other representing the resistance to energy transfer between the interface and the main gas body. The letter resistance would be expected to be expressible as a power function of the gas rate. Thus, in equation form:

$$\frac{1}{f_h} = R_L + \frac{1}{KG^n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (25)$$

In the original analysis by London, Mason, and Boelter, it was demonstrated that within the accuracy of the data it was permissible to mask the existence of the resistance R_L, (which is, after all, ordinarily a comparatively small part of the total resistance), in representing f_h directly as a power function of G; *i.e.*,

$$f_h = K'G^{n'} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (26)$$

If this may be done, the effectiveness ratio in Equation (24) at once takes on a simple form. Fig. 6 shows a plot of log $\eta\left(\frac{a}{a_{min}} - 1\right)$ *vs.* log G as a test of this premise. It is there possible to draw straight lines of essentially the same slope through all series of points at a constant water rate for which the data are sufficiently accurate and extensive to establish a trend. The spread in the ordinates of the lines certainly appears to be quite definite and must indicate that either K' is a function of the water rate, or that η is a more complicated function of the water rate than was determined in the original paper, or that perhaps the use of the arithmetic-mean air-stream enthalpy for the calculation of the over-all average conductance for energy transfer and the definition of h_{Lm} in terms of water temperature gives rise to the apparent effect of the water rate noted. The value of $(1 - n')$ obtained from Fig. 6 is 0.6, indicating a magnitude of 0.4 for n' as determined by this method instead of 0.48 as found in the original paper, although the data are not extremely sensitive to small changes in n'. A possible explanation behind the evidence of Fig. 6, assuming that the magnitudes of η employed are correct, is that R_L itself (or, at least, what *appears* to be R_L), is a power function of the gas rate with an exponent of the same order as n or n' and a multiplying constant which depends upon the liquid rate. This would lead to the logical conclusion that the liquid surface turbulence occasioned by the gas stream acting on the surface is every bit as important in influencing the temperature difference between the main liquid body and the interface as the liquid rate itself, which rate determines the average thickness of the liquid film on the packing surface.

A plot of the resistance Equation (25) was tried as an attempt to evaluate R_L as the intercept on the ordinate axis and to determine $1/K$ as the slope for each curve at a constant water rate. The data were too scattered to permit a successful plotting from which accurate quantitative conclusions could be drawn, however.

XUM

TABLE 3—COMPARISON BETWEEN THE OVERALL AVERAGE CONDUCTANCES COMPUTED FROM THE LOGARITHMIC MEAN ENTHALPY POTENTIAL AND FROM THE COOLING DIAGRAM

Run No.	t_1 F	H_1 milli-lb / lb	h_1 Btu / lb dry air	Log mean Δh Btu / lb dry air	f_h, Btu / hr ft² Btu/lb From log mean Δh	$\left(\frac{f_h, \text{log mean}}{f_h, \text{cooling diagram}}\right)$	Average Ratios[a]	L/aV lb / hr ft²
1	79.0	21.6	42.7	13.6	9.14	1.01		8.61
2	77.0	20.0	40.4	13.0	9.66	1.12		...
3	74.8	18.4	38.1	11.7	10.7	1.02		...
4	70.5	15.5	33.9	9.57	14.5	1.13		...
5	69.8	15.2	33.4	10.4	14.4	1.01		...
6	70.0	15.1	33.3	10.3	16.5	1.15	1.07	...
8	78.4	20.9	41.7	12.8	10.9	0.91		10.8
9	76.7	19.7	40.0	9.50	13.8	1.01		...
10	77.5	20.5	41.0	10.5	15.8	0.94		...
11	78.6	20.6	41.4	11.0	15.3	1.01		...
12	73.3	17.6	36.8	8.39	17.3	1.04	0.98	...
13	84.4	25.1	47.8	10.8	12.7	1.12		13.0
14	81.5	22.4	44.1	10.4	13.1	1.07		...
15	80.4	21.7	43.1	11.3	16.1	1.11		...
17	78.8	20.6	41.5	8.40	17.6	0.98		...
18	76.2	18.9	39.0	9.09	19.5	0.95		...
19	76.3	19.6	39.8	8.19	21.9	1.19		...
20	75.2	19.0	38.9	7.96	19.2	1.03	1.06	...
22	78.7	21.2	42.1	12.1	16.7	1.11		15.0
23	80.9	22.4	44.0	10.8	15.3	1.02		...
25	77.6	20.4	41.0	11.9	20.0	0.98		...
26	76.0	19.2	39.3	11.3	19.1	0.94	1.01	...
28	94.9	36.6	63.1	20.3	12.1	0.92		17.3
29	...	...	...	...	...	...		...
30	83.8	25.3	47.9	14.0	13.9	1.01		...
31	81.8	23.2	45.1	13.0	16.7	1.04		...
32	79.9	20.6	41.8	12.9	17.1	0.94		...
33	72.9	16.9	36.0	9.87	20.0	1.08		...
34	72.1	16.1	34.9	9.41	21.0	1.04	1.01	...
35	96.0	37.8	64.7	21.2	12.2	1.07		22.1
36	82.5	24.1	46.3	11.0	13.9	1.10		...
37	84.9	26.2	49.0	15.2	14.7	1.07		...
38	82.2	23.7	45.8	15.3	16.1	0.97		...
39	74.1	18.0	37.5	9.57	17.6	1.04		...
40	80.0	21.8	43.1	12.9	20.2	1.28		...
41	74.9	18.4	38.1	11.6	19.7	1.12		...
42	74.0	17.2	36.6	12.2	20.0	1.03	1.06	...
43	85.3	26.3	49.4	12.8	12.3	1.19		25.9
44	88.3	29.6	53.7	16.1	12.5	1.08		...
45	84.2	25.6	48.4	13.3	13.2	1.11		...
46	81.8	23.6	45.5	13.2	14.6	1.11		...
47	81.7	22.6	44.4	14.5	16.0	1.07		...
48	77.7	20.3	40.7	14.0	16.1	1.06		...
49	80.5	22.7	44.2	15.0	17.2	1.18		...
50	81.5	22.6	44.3	14.5	17.1	0.97		...
51	76.2	19.4	39.5	13.2	17.9	1.03		...
52	75.4	18.9	38.8	12.4	19.3	1.11		...
53	72.6	16.7	35.7	10.9	21.6	1.19	1.10	...
54	88.1	29.3	53.4	16.4	11.7	1.10		30.0
55	82.7	24.2	46.5	12.5	12.4	1.13		...
56	82.2	23.9	46.0	14.9	13.7	1.18		...
57	80.3	22.3	43.7	15.7	14.7	1.13		...
58	80.8	22.9	44.5	18.7	15.5	1.06		...
59	76.5	19.2	39.4	14.6	18.6	1.16		...
60	74.5	17.6	37.1	12.3	20.3	1.15	1.13	...

[a] Average for each constant L/aV.

The effect of calculating the over-all average conductance on the basis of the approximate arithmetical mean for the air-stream enthalpy, as indicated by Equation (20), must next be studied in a quantitative manner, in order that any conclusions to be drawn may be set forth for properly defined conditions.

It was shown in the paper by London, Mason, and Boelter (5), that cooling tower performance could ordinarily be predicted with quite sufficient accuracy by the use of the logarithmic mean of the terminal enthalpy potentials when the system operates within a range of water temperature where an approximate linear relationship prevails between the enthalpy of the saturated air-water vapor mixture and the temperature. Therefore, in order to avoid the graphical work which would otherwise be necessary (9), it would follow that a com-

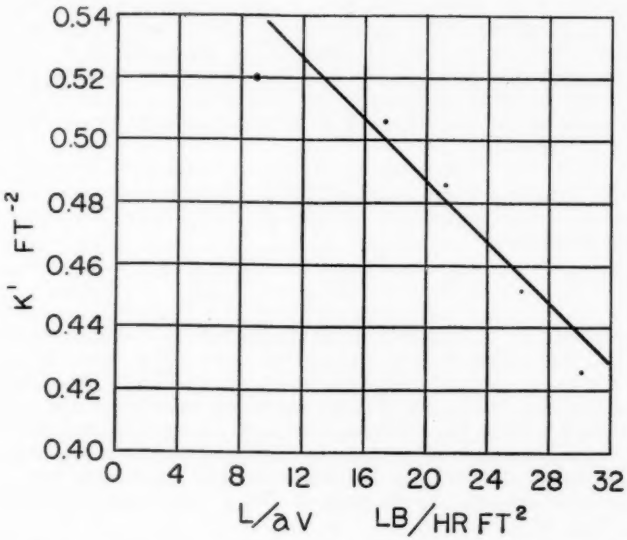

FIG. 7. EFFECT OF THE WATER RATE UPON THE MULTIPLYING CONSTANT K' IN EQUATION (26) FOR THE OVERALL ENERGY CONDUCTANCE

Key

L/aV = 8.61 lb/hr ft²
L/aV = 10.8 lb/hr ft²
L/aV = 13.0 lb/hr ft²
L/aV = 15.0 lb/hr ft²
L/aV = 17.3 lb/hr ft²
L/aV = 22.1 lb/hr ft²
L/aV = 25.9 lb/hr ft²
L/aV = 30.0 lb/hr ft²

parison between magnitudes of f_h determined using the logarithmic-mean enthalpy potential and the corresponding values previously obtained from the cooling diagram would serve to indicate in general to what extent the apparent variation of f_h with the water rate is brought about by the method of analysis.

The results of such a comparative study are shown in Table 3. The trend in the variation of the ratio of the logarithmic-mean conductance to the cooling-diagram conductance appears to be a slight increase with increasing

water rate, but it is not possible to safely draw a more precise quantitative conclusion in view of the limitations prevailing upon the accuracy of the logarithmic mean as a basis of comparison. The fact that magnitudes of the over-all average conductance obtained from the cooling diagram tend to be the lower of the two sets of values would be expected in line with the use of the arithmetic-mean air-stream enthalpy causing the average transfer potential to be a bit greater than presumably it should be. The trend indicated here would also provide further justification for the assumption of numerical equivalence between f_h and f_D for preliminary design purposes.

According to the evidence on hand, it follows that the conclusion to be regarded as prevailing in this question in its present status is that a small but nevertheless apparently very real effect of the water rate upon the over-all average conductances determined by Merkel's method seems evident. However, the apparent variation according to the cooling-diagram analysis is perhaps slightly magnified by the very way in which the cooling factor basic to the diagram is defined.

Returning to Fig. 6, it is possible to compute magnitudes of K' for Equation (26) according to the location of each curve at a constant water rate. The results of such an analysis are plotted in Fig. 7 and are shown in the following tabulation:

L/aV, lb/hr ft³	8.61	17.3	22.1	25.9	30.0
K', ft^{-2}	0.520	0.506	0.486	0.451	0.426

These values for K' are only rough, as the curves in Fig. 6 were merely faired by eye, but they serve to indicate the trend. More extensive and accurate test data are needed to establish the function of Fig. 7 more securely.

It would appear that, as a first rough approximation, the curve of Fig. 7 may be taken as a straight line within the range of the points plotted for complete wetting of the packing. The poor correlation for points at low water rates might be remedied through a redefinition of the fraction of total packing surface wetted according to the curve of Fig. 7. The equation of the faired straight line is:

$$K' = 0.593 - 5.16 \times 10^{-3}(L/aV), \text{ ft}^{-2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (27)$$

Substituting the expression for K' given as Equation (27) into Equation (26), and then replacing f_h in Equation (24) for Merkel's effectiveness by its equivalent so determined, there follows, with $aV = 418$ sq ft for the tower in question,

$$\frac{\alpha_{min}}{\alpha} = \frac{1}{1 + \dfrac{G^{0.6}}{\eta[124 - 1.08(L/aV)]}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (28)$$

The preceding equation may be further reduced to one involving the air and water rates only by substituting for α_{min} its equivalent, $L/2G$, if such a form is desired. It is to be understood that the use of a relationship for K' in the form of Equation (27) is necessary to establish the desired interpolation equation for use in conjunction with the cooling diagram, but the available evidence in its support is at present limited to the particular method of analysis involved, and, therefore, immediate generalization and application to other methods of analysis is not warranted.

With the aid of Equation (28) the cooling diagram could now be employed to study the detailed performance of the tower under varied conditions within the range covered by the data. The present investigation will not be carried beyond this point however, for its purpose has been primarily to develop and explore the feasibility of a method of analysis. In the original article by Merkel are several performance comparisons and studies of performance trends under changing operating conditions which should prove of interest in indicating further possibilities.

A further study in the use of an enthalpy-temperature diagram for the solution of evaporative cooling problems has been presented by Koch (10). The more complicated form of analysis resulting if the simplifying approximations of Merkel's development and the assumption of the Lewis equation are not introduced is there presented. A comparison further substantiates the observation that Merkel's method is quite adequate and accurate so long as excessively high water temperatures and large cooling ranges are avoided.

SUMMARY

A method of analyzing and correlating performance data on air-water evaporative cooling systems, as devised initially by Friedrich Merkel, has been developed and applied to the study of data on a particular forced-draft packed cooling tower. It is believed that the method will prove of value especially because of its simple and direct nature, involving merely the use of a diagram and a minimum of computation. The concept of superimposed heat and mass transfer as representing an energy transfer, and the convenient although slightly approximate simplification possible through the use of the Lewis equation, were basic to the development. It was also demonstrated that for the tower studied there appeared to exist a small but definite resistance to energy transfer from the main liquid body to the liquid-gas interface. A form of empirical interpolation equation for the effectiveness of the evaporative cooling process was established to be used in connection with the cooling diagram.

ACKNOWLEDGMENT

The author wishes to express his appreciation for the helpful suggestions made by Prof. L. M. K. Boelter of the University of California, A. L. London of Stanford University, and R. A. Seban of Stanford University.

NOMENCLATURE

a = packing surface transfer area per unit of tower volume, ft²/ft³
c_a = unit heat capacity at constant pressure of the dry air, Btu/lb F
c_s = unit heat capacity at constant pressure of the water vapor, Btu/lb F
c_L = unit heat capacity at constant pressure of the liquid water, Btu/lb F
f_e = over-all unit surface conductance for convective heat transfer, Btu/hr ft² F
f_D = over-all unit surface conductance for convective mass transfer, lb/hr ft² lb/lb
f_h = over-all unit surface conductance for energy transfer, Btu/hr ft²/Btu/lb
G = dry air flow rate through the system, lb/hr
h = enthalpy of the air-vapor mixture, Btu/lb dry air
h_s = enthalpy of the air-vapor mixture saturated at temperance T, Btu/lb dry air
h_{fg} = latent heat of evaporation of the water, Btu/lb

XUM

H = specific humidity of the air, lb vapor/lb dry air
H_s = specific humidity of the air saturated at temperature T, lb vapor/lb dry air
K, K' = multiplying constants in the expressions for the energy conductances as power functions of the gas rate
L = liquid water flow rate through the system, lb/hr
n, n' = exponents of the gas rate in the expressions for the energy conductances as power functions of the gas rate
NTU = number of transfer units
q_c = rate of convective heat transfer, Btu/hr
q_D = rate of heat transfer from the main water body due to convective mass transfer, Btu/hr
R_L = resistance to energy transfer from the main liquid body to the liquid-gas interface, (Btu/hr ft^2 Btu/lb)$^{-1}$
s = unit heat capacity at constant pressure of the air-vapor mixture, Btu/(lb dry air) F
T = average temperature of the water body at any cross-section, F
t = average temperature of the air-vapor mixture at any cross-section, F
T_o = datum temperature for the measurement of enthalpies, F
V = volume of the energy exchange system, ft^3
α = the cooling factor, defined by Equation (22)
$\lambda = G/L$
η = fraction of the total packing surface wetted

Subscripts

o refers to the datum state
$_m$ refers to a mean value
i refers to interface conditions
$_L$ refers to conditions at the main water body temperature
1 refers to water-inlet and air-outlet conditions
2 refers to water-outlet and air-outlet conditions

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Name	Joined
Albert R. Acheson	1919
Harry Bentz	1919
J. J. Blackmore (*Charter and Life Member*)	1895
Henry B. Gombers (*Life Member*)	1901
Edward B. Gordon, Jr. (*Life Member*)	1908
George G. Harrison	1937
Henry W. Hart-Baker	1918
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Alfred S. Kellogg (*Life Member*)	1916
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Louis J. Sydow	1936
Arthur J. Thom	1939
Maurice I. Weil	1928

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